



Proceedings

ASPE 2016 Annual Meeting
October 23-28, 2016

Portland Marriott Downtown Waterfront
Portland, Oregon, USA

American Society for
Precision Engineering

P.O. Box 10826, Raleigh, NC 27605-0826
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Proceedings of

The Thirty-First Annual Meeting of the American Society for Precision Engineering

October 23-28, 2016
Portland Marriott Downtown Waterfront
Portland, Oregon, USA

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

The Annual Meeting has evolved into an international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

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Preface

This book comprises the proceedings of the 31st ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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ASPE gratefully acknowledges the time and the effort of the organizing and technical program committee to bring the precision engineering community this program.

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ASPE Lifetime Achievement Award

Dr. Stuart T. Smith

University of North Carolina-Charlotte



Dr. Stuart Smith is a Professor of Mechanical Engineering in the Center for Precision Metrology at UNC Charlotte and was cofounder of North Carolina companies Albany Instruments Inc., 2000, Motus Mechanical Inc., 2014, and was involved in the origination of InsituTec Inc. in 2002. These research and manufacturing companies specialize in eddy current sensing technologies, mechanism construction kits and fine motion control metrology systems respectively. He started his career in 1977 when he took up a factory maintenance apprenticeship with Miles Redfern Ltd (UK), a manufacturing industry producing rubber and plastic components for the auto industry. After completing his Ph.D. degree at the University of Warwick he continued working as a faculty member at this institute for 6 years. In 1991 he was awarded an engineering secondment by the Royal Academy of Engineering and subsequently spent a year on sabbatical leave working at the Center for Precision Metrology at the University of North Carolina at Charlotte. In 1992, he returned to Charlotte to take up a permanent position.

Throughout his research career his main focus has been the development of instrumentation and sensor technologies, including advanced signal processing techniques, for measurement of surface profile, micro-geometry and displacements, primarily aimed towards the challenges of atomic scale discrimination and modifications. Development of these systems has required the innovation of many high bandwidth, precision positioning and translation systems. This work has resulted in 18 patents, over 90 journal publications, 90 conference proceedings and the authorship of two books (*Flexures: Elements of elastic mechanism design* and *Foundations of Ultra Precision Mechanism Design*) the first of which remains in print after 20 years. Current interests include the integration of multi-sensing methods into precision electromechanical systems for nanometer and micrometer level assembly, x-ray interferometry and micro CT metrology, dimensional metrology, and the creation of new instrumentation capabilities.

ASPE Distinguished Service Award

Dr. John S. Taylor

Lawrence Livermore National Laboratory
University of North Carolina Charlotte



Dr. John Taylor joined Lawrence Livermore National Laboratory (LLNL) during the commissioning of the Large Optics Diamond Turning Machine, exploring diamond machinability and tool wear. John served as Optics Group Leader and was Program Leader for LLNL's EUV Lithography Program (with Sandia and LBNL), which demonstrated the first EUV stepper-scanner. A rigorous approach to error budgeting was instrumental in designing and constructing the world's first diffraction-limited wide-field EUV lithographic camera, thus establishing the feasibility of lithographic-quality imaging using soft x-rays.

John is an enthusiastic charter member of ASPE, and served as a Director-at-Large, Secretary, Vice President and President of the Society. Among his most rewarding professional experiences, John has chaired many ASPE Topical Meetings, and is currently chairing the 31st ASPE Annual Meeting. In addition to the many ASPE positions held by John, he developed, implemented and manages the ASPE Technical Leadership Committees. John is a member of ASME, OSA, and is a Fellow of SPIE. He received his PhD in Mechanical Engineering from Purdue University in 1984.

2016 Scholarship Recipients

National Science Foundation Grantees



Students who are studying at United States Institutions are supported by the National Science Foundation under Grant No 1644838 to attend the 31st Annual Meeting of the American Society for Precision Engineering. Support is in the form of the conference registration fee.

ASPE is grateful to the National Science Foundation for its support of students to attend the 31st Annual Meeting of the American Society for Precision Engineering. This support provides student participants with opportunities to exchange ideas with leading researchers and engineers from worldwide industrial and governmental facilities, as well as with students and faculty from both domestic and international universities. Student participants will learn the latest research areas and results in precision engineering fields of interest. These opportunities will not only improve the students' knowledge in the field, but also develop scientists and engineers who are ideally suited to create the next generation of designs in optics, metrology and VSLI manufacturing that will push manufacturing and measurement capabilities.

Carl Zeiss Student Scholarship



The winner of the 2016 Carl Zeiss Student Scholarship is Award is **Timothy S. Ryan**, University of Pittsburgh (Jeffrey S. Viperman, Advisor). This award includes the ASPE Annual Meeting registration fee, 4 tutorial fees, travel expenses and lodging to the Annual Meeting. In addition, the scholarship provides for travel and lodging to Maple Grove, Minnesota to visit Carl Zeiss Industrial Metrology. Mr. Ryan is the fourth recipient of the Carl Zeiss Student Scholarship awarded through ASPE.

ASPE Student Scholarships

ASPE is proud to award 2 student scholarships for the 31st ASPE Annual Meeting in Portland, Oregon. This year's scholarship recipients are **Hossein Mohammadi**, Western Michigan University (John A. Patten, Advisor) and **Mark A. Rubeo**, University of North Carolina-Charlotte (Tony L. Schmitz, Advisor). The ASPE Student Scholarships include a waiver of the Annual Meeting registration fee and 4 tutorial fees. It also includes an honorarium to cover travel and lodging expenses to the 31st ASPE Annual Meeting in Portland. The ASPE Student Scholarships were made possible by **Corning Incorporated, Lion Precision, Moore Nanotechnology** and **many contributing members**.

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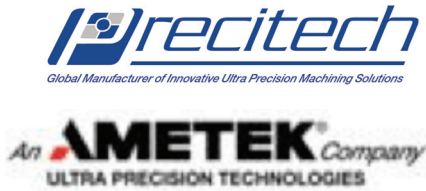
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Technical and Poster Sessions

The technical program of the 31st ASPE Annual Meeting contains over 120 papers on precision engineering state-of-the-art research. Papers that lent themselves to significant verbal interaction have been selected for poster presentation. Authors will present their work during both poster sessions on Tuesday, October 25 from 3:30PM – 5:00PM and on Thursday, October 27 from 3:00PM – 4:00PM.

Welcome Remarks & Session I

Precision Design

Tuesday, October 25, 2016, 8:15AM - 10:00 AM

Session Chair: Mark A. Stocker (Cranfield Precision, Division of Fives Landis Ltd.) and Mark T. Kosmowski (Electro Scientific Industries, Inc.)

1. Precision Design*

Stocker, M. A. (Cranfield Precision, Division of Fives Landis Ltd.);
Kosmowski, M. T. (Electro Scientific Industries, Inc.)

2. The Design and Development of a High Precision Machine for Grinding Large Optics

Fitzpatrick, J. J.; Miller, G. J.; Combrinck, H.; Stocker, M. A.
(Cranfield Precision, Division of Fives Landis Ltd.) 15

3. Stiffness Consequences of Misalignments in an Overconstrained Flexure Design

Nijenhuis, M.; Brouwer, D. M. (University of Twente) 21

4. Mechanical Design of a Sample Manipulator for X-Ray Coherent Surface Scattering Imaging with Flexure Stages for 10-NM-Scale and 10-NRAD-Scale Positioning

Shu, D.; Kearney, S.; Anton, J.; Jiang, Z.; Sun, T.; Maser, J.; Sandy, A. R.;
Wang, J. (Argonne National Laboratory) 26

5. An Ultra High Accuracy 1D Laser-Based Micrometer for Diameter Measurement

Stoup, J. R. (National Institute of Standards & Technology) 30

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Precision Manufacturing

Tuesday, October 25, 2016, 10:30 AM - 12:00 PM

Session Chair: Deepak Ravindra (Micro-Lam Technologies, LLC) and John C. Ziegert (University of North Carolina - Charlotte) and Bradley H. Jared, Sandia National Laboratories

1. Current Advances in Precision Manufacturing: A Review of Selected Areas*

Ravindra D. (Micro-LAM Technologies, LLC); Ziegert, J. C.
(University of North Carolina -Charlotte);
Cheung, C.-F. (The Hong Kong Polytechnic University); Marsh, E. R.
(The Pennsylvania State University); Jared, B. H. (Sandia National Laboratories);
Roblee J. W.; Hashimoto, T.; (AMETEK - Precitech, Inc.)

***No Abstract Available**

2. Enhanced Bidirectional Fabrication of Right Triangular Prismatic Retroreflectors	
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4. A Compact Spectroscopic System Based on Custom-Printed Concave Gratings	
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Kushnir, E.; Clark, W. H. (Hardinge, Inc.); Aguilar, A. (Corning, Inc.)	52

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Reducing the Cost of Precision

Tuesday, October 25, 2016, 1:30 PM - 3:00 PM

Session Chair: Alexander H. Slocum (Massachusetts Institute of Technology) and
Richard M. Seugling (Lawrence Livermore National Laboratory)

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Slocum, A. H.; Teo, D. T. J. (Massachusetts Institute of Technology)	56
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Sun, R.; Li, L. (Washington State University)	78

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Controls and Mechatronics

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Session Chair: Dannis M. Brouwer (University of Twente) and Stephen J. Ludwick
(Aerotech, Inc.)

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3. Non-Collocated Optimal Control of Flexible Drives with Load Side Feedback	
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Session Chair: Robert M. Panas (Lawrence Livermore National Laboratory)
and Michael A. Cullinan (The University of Texas at Austin)

1. Micro-Nano TLC Overview of Research in Precision Micro- and Nano-Technology	
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Session Chair: Christopher J. Evans (University of North Carolina - Charlotte) and Peter J. de Groot (Zygo Corporation)

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*No Abstract Available

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Technical Papers

THE DESIGN AND DEVELOPMENT OF A HIGH PRECISION MACHINE FOR GRINDING LARGE OPTICS

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INTRODUCTION

Cranfield Precision has recently developed a system for machining large complex optical components. This paper describes the design principles used to address the stiffness, stability and dynamic performance requirements necessary for high volumetric material removal and component accuracy during an extended grinding cycle.

In 1989 Cranfield Precision supplied a 2.5m Off Axis Grinding Machine (OAGM2500) to Eastman Kodak in the USA [1]. The configuration of axes used in this latest development provides a higher performance and removal rate within a significantly smaller machine footprint.

MACHINE DESIGN CONCEPT

The machine has two hydrostatic bearing linear axes and one hydrostatic rotary axis. The component is mounted on the table of the rotary C axis which itself is mounted within the linear X axis carriage. By combining the horizontal linear X axis motion with the rotary motion of the C axis, a linear Y axis becomes unnecessary for the production of both on and off axis freeform surfaces.



FIGURE 1. OGM1200.

The hydrostatic grinding spindle is mounted with up to 30° of inclination directly beneath the Z axis carriage. The spindle axis of rotation is parallel with the X axis in the horizontal plane.

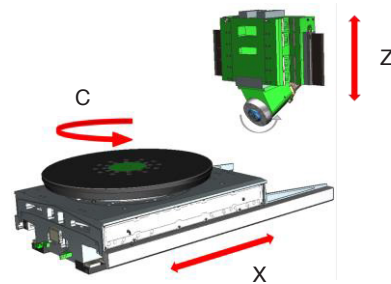


FIGURE 2. Primary C,X and Z axes

The key axis is the vertical linear tool in-feed axis (Z). The position bandwidth of this axis is critical as it directly affects the accuracy of the component surface. Cranfield Precision's unique design used for compensating the Z axis mass contributed significantly to the elimination of the inherent hysteresis error seen at axis reversal when traditional methods are used. This hysteresis error would normally be imparted onto the component surface unless significant reductions in feedrate and accelerations are introduced, with the subsequent increase in process time.

A full description of the patented counterforce mechanism can be read in the extended abstract 4753 A New Improved Counterforce Mechanism for Vertical Machine Tool Axes [2]

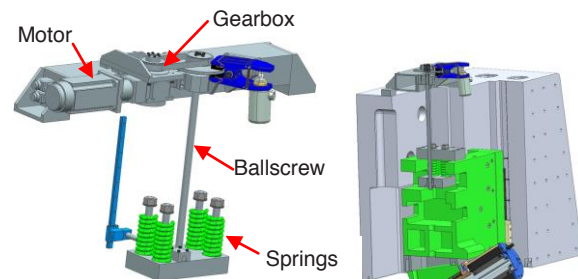


FIGURE 3. Cranfield Precision's counterforce mechanism incorporated into the OGM1200.

OPTIMISING THE DESIGN

Primary forces of mass, acceleration and process have the potential to distort a structure as it reacts. These structural distortions can lead to component inaccuracies and a degradation of surface finish. The design of the OGM1200 structure focused on ensuring that, wherever possible, these primary forces were constrained in such a way that distortion within it was minimized.

Aligning the Forces

The bridge structure that carries the Z carriage and grinding spindle had to resist most of these forces. The 800kg combined moving mass of the Z axis was required to accelerate up to 1m/s^2 at the same time as having to apply process force. Two linear motors were positioned equi-distant either side of the grinding contact point and in approximately the same Y axis plane. This kept the forces through the carriage balanced avoiding any tendency for the carriage to yaw. The ballscrew of the counterforce mechanism was also aligned with the centre of gravity of the Z carriage for the same reason.

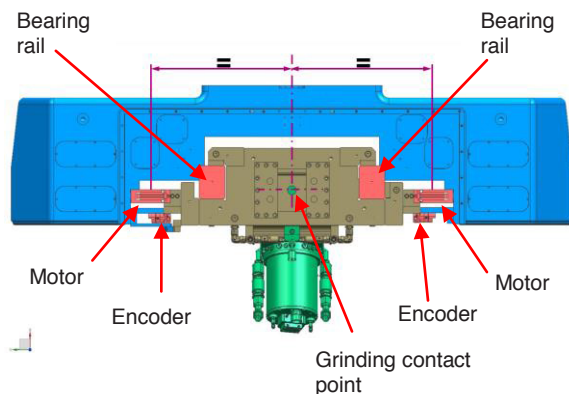


FIGURE 4. Alignment of Z axis components for minimal offset.

The guide rails for the hydrostatic bearings were positioned approximately equi-distant of the grinding contact point in the Y plane to prevent the reaction forces creating a pitch movement in the Z carriage.

The Abbé error between the twin encoders and the grinding contact point was minimized by placing them as close to the grinding contact point in the Y plane as possible.

Balance of the grinding wheel is an important aspect in achieving good surface finish and low

sub-surface damage. Provision was made in the grinding wheel design to facilitate on-spindle balancing down to G0.01 (equates to $<0.05\mu\text{m}$ displacement). However, it was important to ensure that the imbalance forces that would have a direct influence on the surface were constrained by the Z in-feed axis servo.

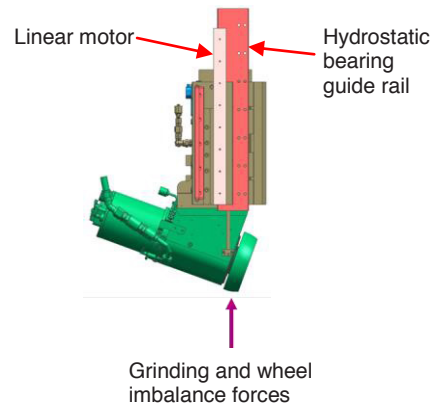


FIGURE 5. Process and imbalance forces in line with Z linear motor

Structural Design

Providing a stiff structure for the forces to react against is essential not only to maintain accuracy and surface finish but also to minimize sub-surface damage.

The fixed part of the linear motor had to be rigidly mounted as it has to resist the forces in the Z axis. By mounting both motors as close to the vertical columns of the bridge structure as possible the reaction force was transmitted almost directly through the columns minimizing the bending moment. The mounting points for the counterforce mechanism that bears the mass of the Z axis follow the same principle.

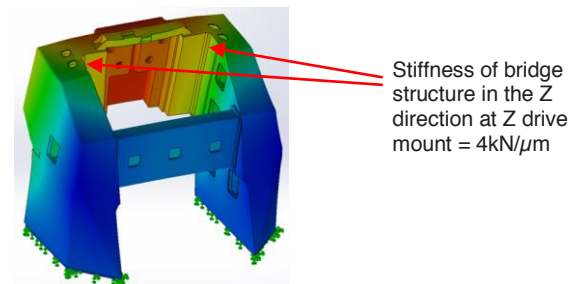


FIGURE 6. Z forces reacting through the bridge structure.

The rigid path of the reaction to the process and acceleration forces is continued down through to the base. The cast iron base is supported on three passive supports at the points of least deflection, two of which by design, are positioned directly underneath the bridge columns.

Whilst the moving mass of the Z axis had little or no distortional effect on the base the 3500kg moving mass of the combined X and C axes had potential to cause significant deformations. Finite Element Analysis was used to design a rigid base structure that could minimize the effects of the X and C axes moving over the 1430mm of stroke.

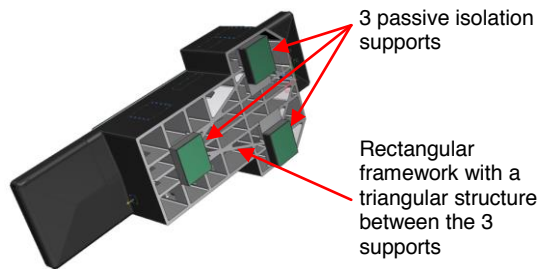


FIGURE 7. Base structure

Analysis of the structure predicted that the change in shape would be $<7\mu\text{m}$ over the complete stroke length of the X carriage. Most of the distortion is in the guiderails rather than the base structure itself.

THERMAL STABILITY

The grinding process time for very large optics can exceed 10 hours and therefore it is essential that the machine design be inherently stable.

The predictable thermal disturbances of the heat created by oil shear in the hydrostatic bearings are managed by controlling the bearing in-feed oil temperature depending on the linear/rotational speed of the bearing to within $\pm 0.1^\circ\text{C}$. The in-feed oil temperature is suppressed so that the oil exiting from the linear and rotary C bearings that returns to tank after flowing over the top surface of the base is kept at ambient temperature $\pm 0.1^\circ\text{C}$. The process coolant is also supplied to the work zone at ambient $\pm 0.1^\circ\text{C}$. By controlling the temperature differential between the machine and the environment thermal distortion is kept to a minimum.

Thermal analysis of the machine structure identified a section that was outside of the machine's metrology loop and was therefore a potential source of thermally induced errors.

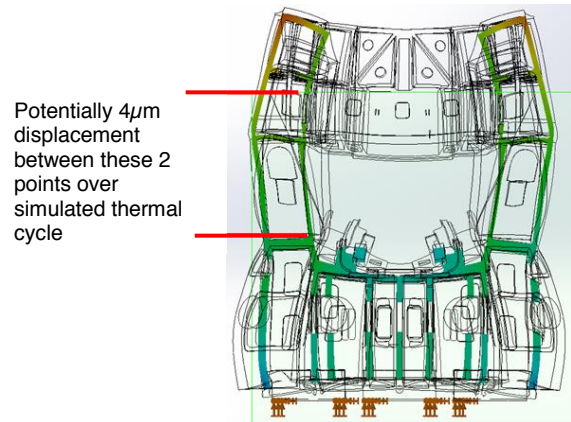


FIGURE 8. Thermal analysis showing the potential path of thermally induced errors.

To minimize this, Invar reference bars were incorporated into the machine structure, providing feedback on any thermally induced movement in critical areas that are outside the axis position loop. This information can be used to compensate the Z axis position.

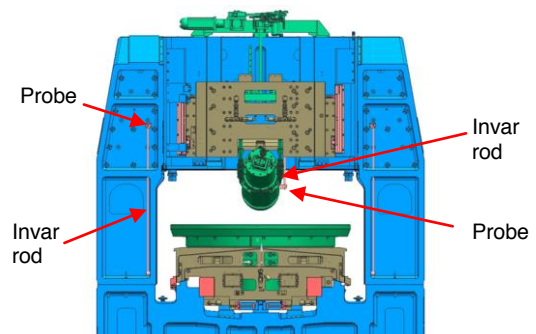


FIGURE 9. Vertical section through bridge showing the invar rods incorporated to provide feedback on any thermally induced structural change.

COMPONENT ROTATION AND POSITIONING

The machine was designed to be able to produce on-axis, off-axis and freeform components. This required the rotary C axis to have not only good rotational accuracy for on axis components but also good positional accuracy for freeform components.

Hydrostatic oil bearings provide high rotational and thrust stiffness with low friction. The higher the number of radial pockets the more the radial error is reduced [3], and within the space available we were able to design a 9 pocket radial and thrust bearing. With a rotating mass including the 1200mm table top, shaft and motor rotor of 980kg the following spindle errors were measured at 60rpm:

TABLE 1. Rotational error analysis of the C axis at 85mm and 285mm above the table surface

Height above table top - mm	Radial error - μm	Axial Error - μm
85	0.471	0.336
285	0.786	0.310

Any variation in the rotational speed of the part would be imparted into the component surface as an error therefore a high torque motor with extremely low torque ripple is directly mounted to the spindle shaft. The encoder rotor is mounted directly to the table top. This enables easy access for the encoder setup whilst ensuring a stiff structural loop for the servo and positional accuracy.

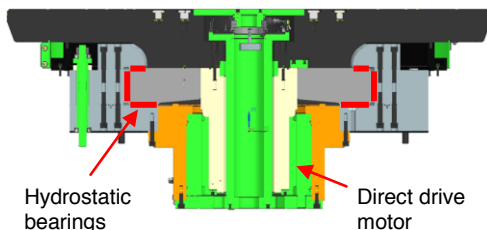


FIGURE 10. Rotary C Axis

COMPONENT METROLOGY

The design of the OGM1200 incorporates a load/unload position that provides clear access from above and the sides for component manipulation. It is also feasible for independent metrology equipment to take advantage of this to perform measurements without removing the component from the machine.

However measurement of large optics can take an extremely long time adding to the overall manufacturing time. An on-machine metrology facility has been mounted to the Z axis allowing the machined surface to be probed at various stages of production providing feedback for path correction due to any process errors or wheel wear etc. The result is that the number of time

consuming full component measurements are reduced.

The probe is lowered pneumatically and held in a kinematic location under gravity. It can then be used in either of two modes depending on the profile of the part surface. If the sagittal depth of the profile is within the range of the probe travel a direct continuous measurement can be taken by nulling the probe using the Z axis and traversing the part underneath the probe using the X axis. For surfaces with a larger sagittal depth the probe can be used as a comparator by zeroing the probe with the Z axis and then following the programmed path with the Z and X axes. The deviation will be the profile error. As the probe uses the same axes and programmed path as the grinding wheel there is a need to qualify the probe path to ensure any inherent errors are excluded from the measurement.

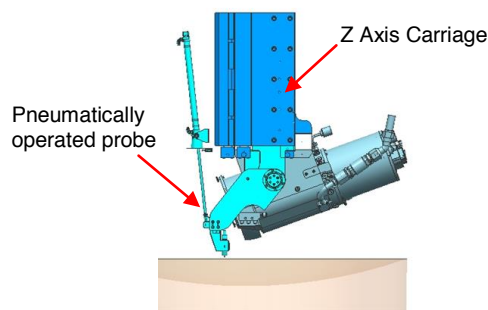


FIGURE 11. On machine metrology mechanism

If required the complexity of the measured data can be increased until ultimately measurements taken at various angles of the C axis can be stitched together to provide a graphical representation of the surface.

MACHINE PERFORMANCE RESULTS

Concave Test Grind

The following data show the results of test grinds designed to test both the thermal stability of the machine and the dynamic response of the vertical, counterbalanced linear axis.

Four 300mm diameter components were ground with a concave sphere of 280mm radius. Each component was ground using a different grinding mode.

1. On-axis sphere grind: C axis 5-50 rev/min (C axis speed increasing as the wheel moves to the centre).
2. On-axis sphere grind: C axis 20 rev/min constant speed.
3. The component was mounted with its axis 0.5mm off-axis from the machine's rotary C axis (i.e. freeform grind path to produce a sphere): C axis 20 rev/min constant speed.
4. The component was mounted with its axis 1.0mm off-axis from the machine's rotary C axis (i.e. freeform grind path to produce a sphere): C axis 20 rev/min constant speed.

The Form Talysurf measurement data over the centre 120mm of the component shows the repeatability of the machine's ground surfaces using the four different grinding modes, demonstrating the machine's repeatability and the effectiveness of the vertical counterbalance system to enable a high position bandwidth of the vertical Z axis.

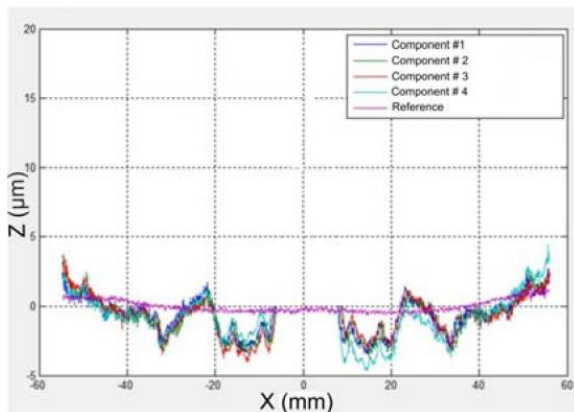


FIGURE 12. Measurement data plots (with unground centre data removed), superimposed upon the reference sphere measurement plot.

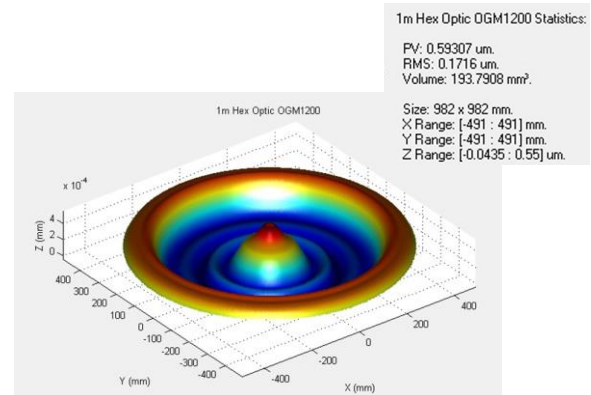
No error correction was employed during the grinding of these surfaces. The primary source of the measured form errors is the errors in the grinding wheel form.

1m Hexagonal with 69m Radius of Curvature Test Grind

To demonstrate the effectiveness of the on-machine metrology to provide feedback, and ultimately compensation, for profile errors due to wheel form and wear, a 1m hexagonal component with a 69m RoC was ground. The

profile was measured after the initial cut and compensation applied. After the final cut measurements were taken with the probe at various C axis angular positions and the results stitched together.

Following the profile with the nulled probe the results show the error to be $<0.6\mu\text{m}$ over the complete surface.



Z Axis Servo Performance

As discussed earlier the critical axis of the OGM1200 is the Z axis and Cranfield Precision's unique counterforce mechanism is a major contributor to achieving the required high performance. The quality of the Z axis servo directly affects the accuracy and quality of the resultant surface.

The performance of the final Z axis was demonstrated by grinding a "saddle" form in to a 280mm diameter part. Vertical sagittal undulations of 15mm at the outer edge gradually decreased towards the centre. With two undulations per revolution and a C axis rotational speed of 25 rev/min, the 800kg Z axis was required to oscillate at 0.84Hz. As can be seen in Figure 13 the position following errors do not exceed $\pm 0.5\mu\text{m}$.

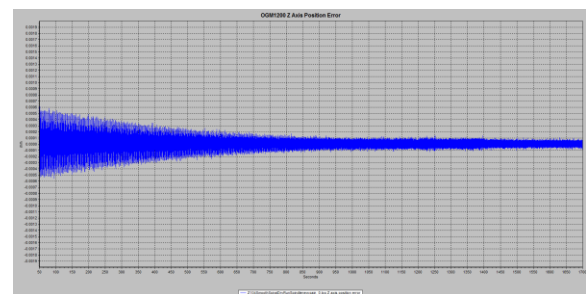


FIGURE 13. Z Axis dynamic performance

SUMMARY

The novel configuration using the minimum number of required axes, reducing force offsets, optimizing structural stiffness and thermal control techniques has resulted in the OGM1200 attaining the stability required by a machine to grind large optics.

Using this platform to apply the latest servo control techniques, test grinds have demonstrated sub-micron repeatability for on-axis, off-axis and freeform grinding modes.

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STIFFNESS CONSEQUENCES OF MISALIGNMENTS IN AN OVERCONSTRAINED FLEXURE DESIGN

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MOTIVATION

Given the drawbacks of exact-constraint design in flexure mechanisms, we seek to understand the behavior of overconstrained flexure mechanisms, such that the limits of operation can be quantified. This potentially enables the use of overconstrained designs for their beneficial properties, such as increased support stiffness [1, 2].

HIGHLIGHTS

1. This work presents *experimental measurements* of the first and two higher eigenfrequencies of an overconstrained two-flexure cross-hinge mechanism set-up. The desktop-size set-up is optimized for stiffness and has a variable misalignment to control the overconstraint stress that affects the systems behavior. The measurements indicate that the both the actuation and support stiffness of the mechanism decrease significantly with increasing misalignment.
2. This work presents *numeric simulations* of the systems behavior that corroborate the experimental measurements. In a sensitivity analysis, several non-ideal effects are identified that explain the deviation between measurement and simulation.
3. This work presents an *analytical prediction* of the bifurcation buckling load due to overconstraint stress. This represents the theoretical critical misalignment above which stiffness is lost.
4. It is observed in this investigation that the *practical* critical misalignment is considerably higher than the *theoretical* buckling limit, indicating that unintended sources of compliance occurring in practice increase the misalignment tolerance in this overconstrained design.

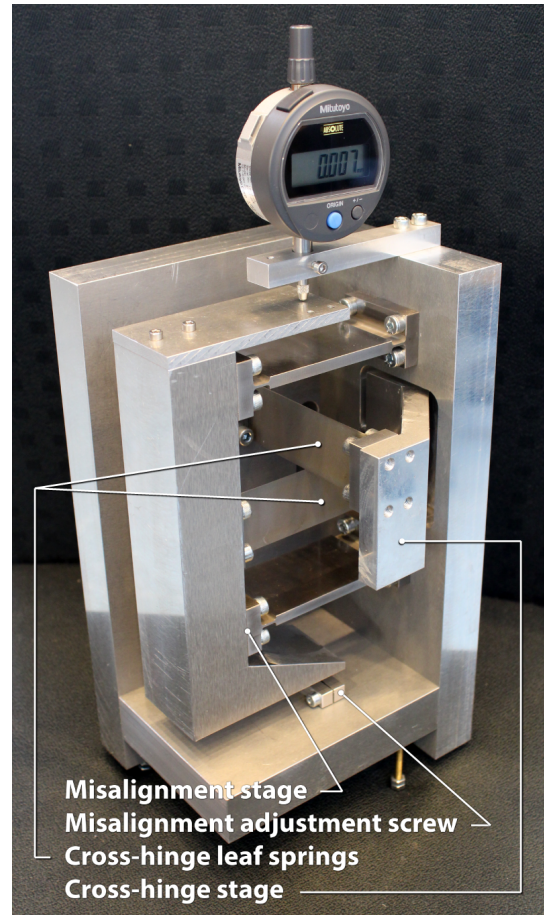


FIGURE 1. Experimental measurement set-up.

INTRODUCTION

The exact-constraint design principle is applied to flexure mechanisms to ensure deterministic behavior. Such designs are not sensitive to misalignments, stemming from e.g. manufacturing tolerances, assembly and temperature gradients. Drawbacks are reduced robustness, support stiffness, load-carrying capacity, and often increased complexity of designs. To explore the potential benefit of overconstrained design in flexure mechanisms, an elementary two-flexure cross-hinge serves as a case study to gain insight.

EXPERIMENTAL SET-UP

The experimental set-up (Figure 1) is based on a cross-hinge with two leaf springs (dimensions $80 \text{ mm} \times 30 \text{ mm} \times 0.35 \text{ mm}$ and crossing at right angles). This design is overconstrained in the translational direction of the rotation axis. The two leaf spring ends are connected to a freely moving rigid stage, the third end is connected to the base frame, and the fourth end is connected to a translation stage for setting the misalignment displacement in the overconstrained direction. On one side of the translation stage, a fine-thread screw is used to provide the overconstraint force; the corresponding misalignment is measured on the other end to avoid unwanted translation stage compliance in the measurement. The cross-hinge stage motion is measured with a laser displacement sensor ($5 \mu\text{m}$ accuracy). Due to the vertically-oriented rotation axis of the cross-hinge, the effects of gravity work mainly in the overconstrained direction of the mechanism, and not in the actuation direction.

To reliably control misalignments in the micrometer range, the set-up has several design features that reduce parasitic compliance and hysteresis:

- The load-carrying frame parts are dimensioned to have very high stiffness.
- The cross-hinge design consists of only two leaf springs, to have a minimal number of parts. This way parasitic compliance due to interface connections is minimized.
- The translation stage is a flexure mechanism itself, to increase repeatability. The support stiffness is very high; the stroke is limited to only 1 mm (before yielding).
- Each leaf spring with the two support fillets is made of a single piece of material (stainless steel, AISI 420) by wire EDM. This minimizes the consequences of difference in strain between support fillet and leaf spring.

EXPERIMENTAL MEASUREMENTS

From measurements of the free oscillation (due to a small initial nonzero displacement) of the cross-hinge stage, the first eigenfrequency can be calculated. It has relevance as it is directly related to the actuation stiffness of the cross-hinge mechanism. Figure 2 shows that the eigenfrequency decreases for both a positive and a negative misalignment setting, from a nominal high value of

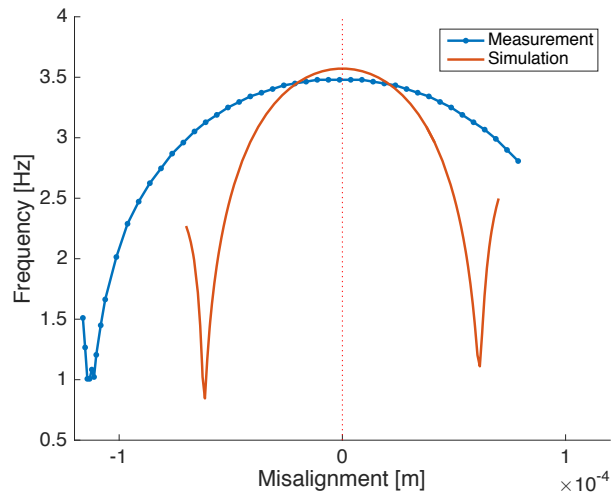


FIGURE 2. First natural frequency.

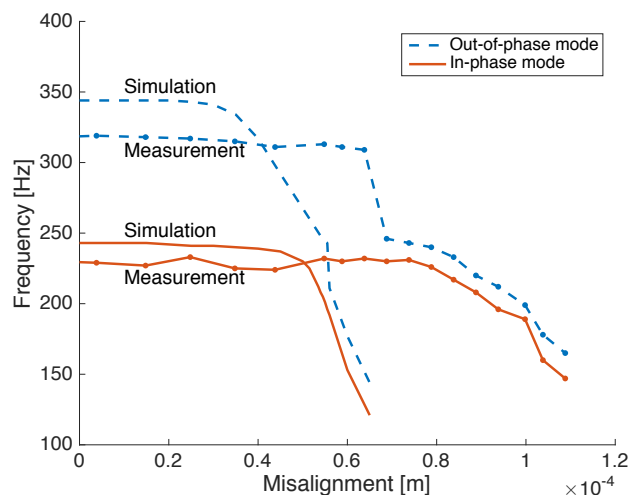


FIGURE 3. Higher natural frequencies.

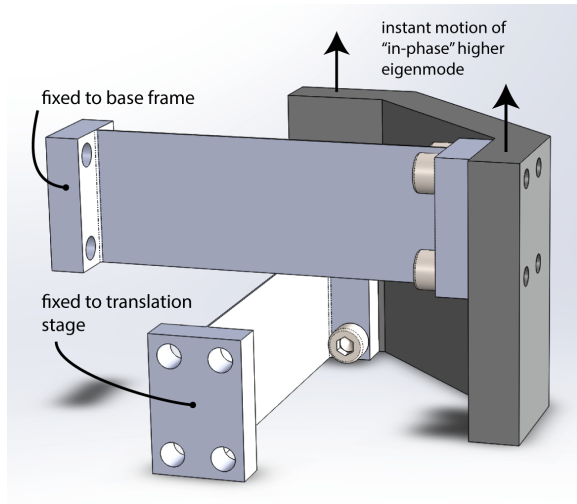


FIGURE 4. Visualization of higher eigenmodes.

3.48 Hz. In the negative misalignment range, a minimum of 1.0 Hz is obtained for $-113 \mu\text{m}$. This indicates a nearly buckled state, at which most of the actuation stiffness is lost. While the behavior for the positive misalignment range seems to mirror the negative, the unstable buckled state is not reached because the hinge stage visibly rotates towards a new stable equilibrium configuration, away from the initial configuration, before the critical misalignment level is reached. The difference for negative and positive misalignments highlights the extreme sensitivity of the buckling phenomenon to small practical imperfections.

Over the positive misalignment range, two higher eigenmodes have been measured by means of impact modal testing. These modes correspond to rigid body motion of the cross-hinge stage orthogonal to the actuation mode. For visualization, Figure 4 shows a cross-hinge schematic (without frame parts) with the instantaneous motion vectors of the stage that correspond to the so-called “in-phase” higher eigenmode. In the other higher eigenmode, the “out-of-phase” mode, one of the motion vectors has a reversed sign. Figure 3 shows that the eigenfrequencies associated with the two higher eigenmodes also decrease significantly (with factors 1.9 and 1.6, resp.) with increasing misalignment. For these higher modes, the decreasing eigenfrequencies indicate that misalignments cause a reduction of the support stiffness of the mechanism, and – hence – a decrease in mechanism performance. Compared to the actuation direction in which an actuator could compensate for stiffness loss, a reduction of support stiffness generally has more

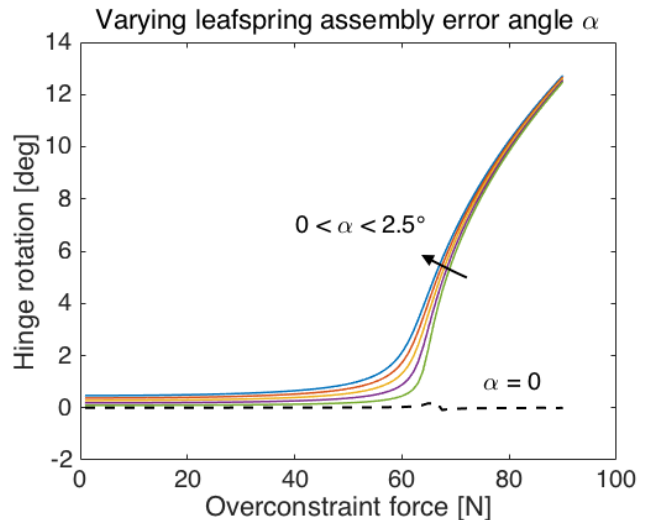


FIGURE 5. Unwanted hinge rotation due to leafspring assembly error.

severe consequences.

NUMERIC SIMULATIONS

The system is modeled with the flexible multi-body software SPACAR (suited for flexure mechanisms [1]) using two-node beam elements [3]. The simulation takes into account effects of gravity and constrained warping at the leaf spring ends, and is capable of calculating stable and unstable (buckled) equilibrium configuration due to input displacements and forces. From a numeric modal analysis, eigenfrequencies are obtained for varying misalignment levels, depicted in Figures 2 and 3.

It is observed that the simulation matches well with the measurements in the unstressed (zero misalignment) case. For nonzero misalignment, the behavior matches qualitatively. Both the first and higher eigenfrequencies drop with increasing misalignment. The simulated first eigenmode clearly shows the buckling phenomenon for positive and negative misalignment. The important difference between measurement and simulation is the critical misalignment level at which buckling occurs: it is considerably lower in the simulation than in the measurement ($62 \mu\text{m}$ versus $113 \mu\text{m}$). The simulation apparently overpredicts the stiffness, most likely due to unmodeled sources of compliance stemming from geometric imperfections and non-ideal boundary conditions in the real set-up.

A sensitivity analysis shows that the discrepancy can be fully accounted for by e.g. a leaf spring as-

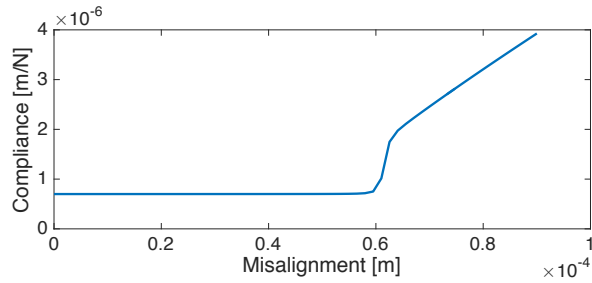


FIGURE 6. Compliance of the cross-hinge stage in vertical direction.

sembly error of only 2.5 degrees about its longitudinal axis, and a tilt angle of the entire mechanism of 2.2 degrees with respect to the gravity vector. While such imperfections are difficult to measure in practice, the stage rotation that occurs as a result of the imperfections upon reaching the buckling load (Figure 5) is observed in the experiment.

Analysis of these potential error sources also conveys that pure bifurcation buckling — in which the first eigenfrequency actually becomes zero and all actuation stiffness is lost — only occurs in the idealized case, i.e. without any effects of gravity and mechanism assembly errors. To illustrate, a gravity component of $0.001g$ in the horizontal plane creates a slight nonzero stage rotation equilibrium configuration and already increases the minimum first eigenfrequency at the critical misalignment level from 0 to 1.0 Hz (which is the value observed in the measurements).

While the modal analysis of the experiment set-up yields eigenfrequencies that serve as a convenient proxy quantity for obtaining stiffness properties of the set-up, the simulation software directly calculates compliances. Figure 6 shows the compliance of the motion stage in the vertical direction. It conveys that there is a clear critical misalignment level at which the compliance suddenly increases, (and keeps increasing with misalignment — while at the same time the stage is rotating), compromising the mechanism's performance.

ANALYTICAL PREDICTION

In literature, several expressions are available for the critical load that induces lateral-torsion bifurcation when bending a beam in the plane of greatest flexural rigidity (neglecting gravity, warping, shear and elongation effects). For a beam of length L , flexural rigidity EI_z and torsional rigidity GJ , the critical load P_{cr} (applied at the beam end)

is given by

$$P_{cr} = \gamma \frac{\sqrt{EI_z GJ}}{L^2},$$

where factor γ depends on the boundary conditions. For a fixed – free beam, $\gamma \approx 4.013$ [4]. For a fixed – fixed beam (meaning that lateral bending is prevented at both sides), $\gamma \approx 2.2\pi$ [5]. These values can be used to estimate bounds for the two-leaf spring system of the current investigation; the critical buckling force lies between 42.0 N and 72.4 N, depending on the boundary conditions where the two leaf springs are attached to the motion stage of the cross-hinge. The buckling force calculated numerically with SPACAR is 59.2 N and lies in between the two analytical values. This suggests that the motion stage does not entirely constrain or completely allow lateral bending of a leaf spring, which seems reasonable.

For a more detailed investigation of the buckling load of the current system, it is suggested to treat both leaf springs simultaneously and properly capture the boundary condition at the motion stage, similar to the analysis method of Meijaard et al. [1].

CONCLUSION

For a simple overconstrained design, it has been shown that the critical misalignment level at which support and actuation stiffness is lost, is considerably larger in practice than the idealized theoretical value. Also, the observed decrease in stiffness is smaller. Practical mechanisms seem more tolerant to misalignment errors than the 'ideal' bifurcation buckling load suggests, due to extra sources of compliance that are present.

ACKNOWLEDGMENTS

We thank Mr. L. Tiemersma for building the experimental set-up, and Mr. Z.A.J. Lok for providing the impact modal testing equipment.

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MECHANICAL DESIGN OF A SAMPLE MANIPULATOR FOR X-RAY COHERENT SURFACE SCATTERING IMAGING WITH FLEXURE STAGES FOR 10-NM-SCALE AND 10-NRAD-SCALE POSITIONING

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INTRODUCTION

The Advanced Photon Source (APS) at Argonne National Laboratory (ANL) is a national user facility for synchrotron radiation research. The APS Upgrade's massive increase in coherence [1] will particularly benefit several specific types of x-ray science techniques, including coherent x-ray surface scattering imaging. The development of coherent surface scattering imaging in grazing-incidence geometry takes advantage of enhanced x-ray surface scattering and interference near total external reflection. It is promising wide applications in elucidating structures in substrate-supported and buried 3D nanopatterns, nanoelectronics, and photonics [2]. However, an extremely stable sample manipulator with ultra-high positioning resolution is needed for this new x-ray technique. In addition, a 100-nm-level sphere of confusion of the rotation stages is required. Therefore reproducible rotations are critical for correcting the eccentric motion possibly introduced by the stages.

GENERAL ASSEMBLE OF THE APS Z4-41 PRECISION SAMPLE MANIPULATOR

To meet the challenging requirements of sample handling in the grazing incidence geometry, a prototype of precision sample manipulator APS Z4-41 for coherent surface scattering imaging has been designed and constructed at the APS. As shown in Figure 1, the sample manipulator is a combination of six precision stages: a commercial base rotary stage; a Z4-4101 rotary flexure stage; a Z4-4102 flexure tip-tilting stage; and a customized x-y-z stages group Z4-4103. The Z4-41 precision sample manipulator is mounted on the top of a set of HuberTM goniometer stages group, which provides general multidimensional positioning for the sample.

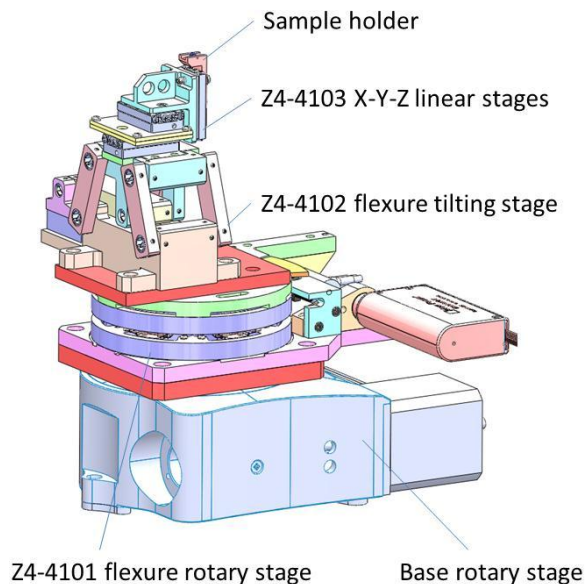


Figure 1. 3-D model of the prototype of APS Z4-41 precision sample manipulator for coherent surface scattering imaging.

APS Z4-4101 COMPACT ROTARY FLEXURE NANOPositioning STAGE WITH TEN-DEGREE-LEVEL TRAVEL RANGE

The sample manipulator uses a PITM MicosTM PRS-110 precision rotary stage [3] as the base rotary stage to position the sample around the main axis with a large angular coverage. Coaxially mounted on the top of the base stage, the APS Z4-4101 rotary flexure stage consists of a pair of rounded-fishbone-shaped flexure modules [4-6]; a stage base with driver mounting flange; a connecting carriage; a stage top carriage with driving arm; and a PicoMotorTM-driven linear actuator [7] combined with a PITM PZT actuator. It provides high angular resolution positioning to ten-nanoradians for the sample main axis with an angular travel range of ~10 degrees. Figure 2 shows a 3-D model of the APS Z4-4101 compact rotary flexure stage.

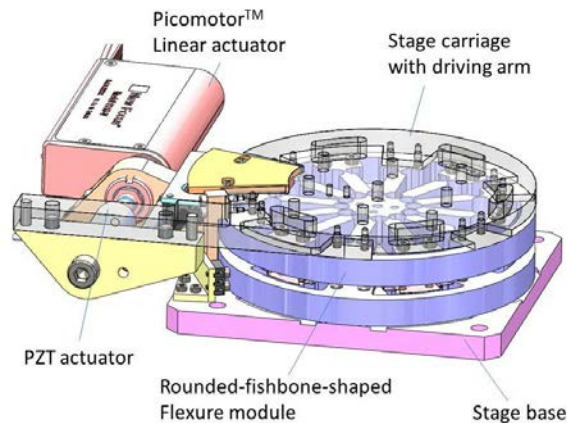


Figure 2. 3-D model of the prototype of the APS Z4-4101 rotary flexure stage.

The rounded-fishbone-shaped module is constructed with stacks of thin metal weak-link sheets which are manufactured using photochemical machining processes with lithography techniques [4,8]. The module is a solid laminar bonded complex structure designed for ultrahigh positioning sensitivity and stability performance.

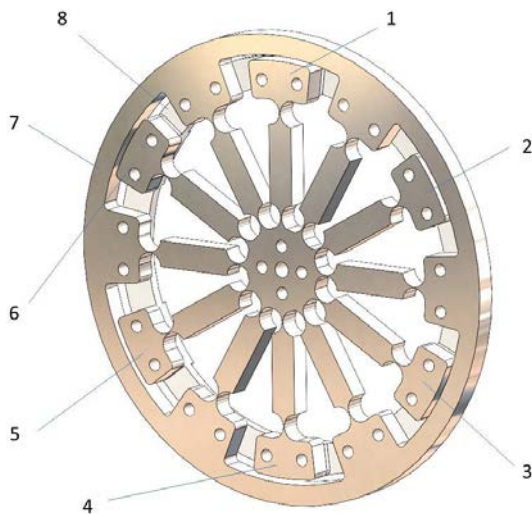


Figure 3. 3-D model of the rounded-fishbone-shaped flexure module assembled with a connecting carriage (item 8). Items 1 to 6 are terminals mounted to the connecting carriage. Item 7 is a ring with terminals mounted to the rotary stage base or the top carriage.

As shown in Figure 3, the items 1 to 6 of the rounded-fishbone-shaped flexure module are terminals mounted to a connecting carriage, item 8. Item 7 is an outer ring with terminals mounted to the rotary stage base or the top carriage of the rotary stage. The outside

diameter of the outer ring, item 7, is 90 mm. Finite element analysis simulated a ~ 3 degree outer ring angular displacement condition while terminals 1 - 6 are fixed as shown in Figure 4. With this load condition, the maximum Von Mises stress reaches ~ 1100 MPa, which is about 92% of the tensile yield strength of the stainless steel 17-7 PH as shown in Figure 5.

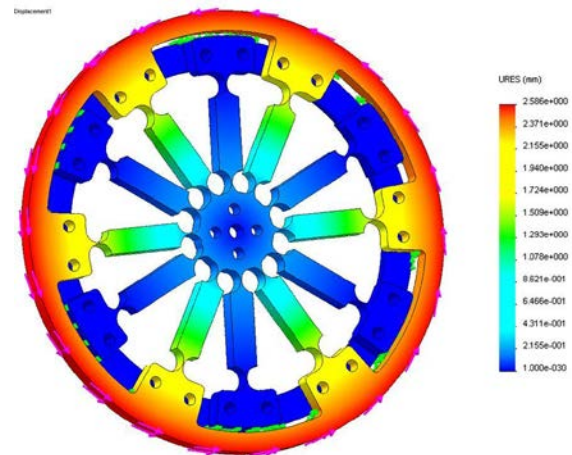


Figure 4. A finite element analysis result for a ~ 3 degree angular displacement of the rounded-fishbone-shaped laminar flexure module for the APS Z4-4101 rotary flexure stage.

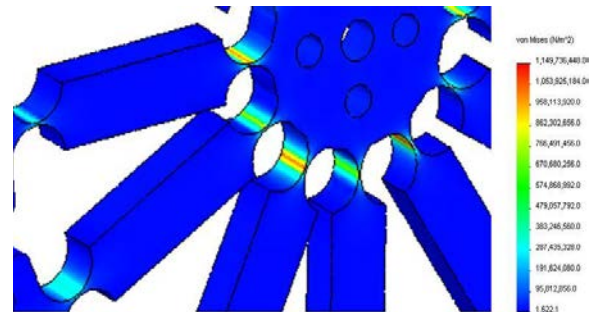


Figure 5. Maximum Von Mises stress reaches ~ 1100 MPa while the flexure module performs a ~ 3 degree angular displacement.

The flexure stage's coarse motion is performed by a Picomotor™ linear actuator. Using a closed-loop Picomotor™ linear actuator as shown in Figure 2, in which a linear grating encoder is integrated, the stage's coarse motion has a positioning repeatability of ~ 3 micro-rad in the entire travel range of 12 degree (with necessary calibration) with a ~ 400 nrad resolution. Similar performance can be reached with the installation of a rotary grating encoder, such as Micro-E™ encoder [9] for stage's coarse motion driven by an open-loop Picomotor™ linear actuator. With the installation of a laser-interferometer system, such as Attocube™ FPS 3010 system [10], the

Z4-4101 stage is designed to have a closed-loop positioning resolution of 10 nrad and repeatability of ~50 nrad for the entire travel range of ~12 degree [6].

APS Z4-4102 COMPACT FLEXURAL TIP-TILTING STAGE

Mounted on the top of the Z4-4101 rotary flexure stage, the Z4-4102 flexural tip-tilting stage is constructed with a 4-bar flexural bearing structure operated by a SmarAct™ PZT-driven linear stage SLC-2430-S-HV [11] to provide a precise angular positioning around the sample's second axis, which is perpendicular to the main axis. The Z4-4102 tip-tilting flexure stage has an angular travel range of 6 degrees with ~0.7 microradian angular positioning resolution. Figure 6 shows a 3-D model of the APS Z4-4102 compact flexural tip-tilting stage.

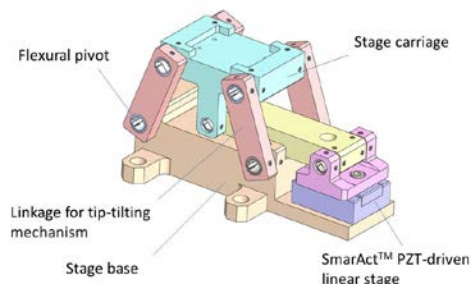


Figure 6. 3-D model of the APS Z4-4102 flexural tip-tilting stage.

CUSTOMIZED X-Y-Z STAGES GROUP

The customized x-y-z stages group Z4-4103 includes three SmarAct™ PZT-driven linear stages SLC-1730-S-HV [8]. Figure 7 shows a 3-D model of the customized stages group. It is designed to manipulate the sample in a 12 mm x 12 mm x 12 mm space with 50 nm closed-loop positioning resolution.

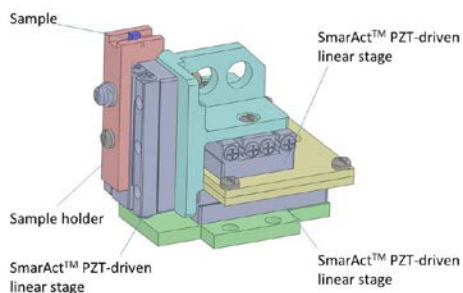


Figure 7. 3-D model of the APS Z4-4103 customized PZT-driven linear X-Y-Z stages group.

PRELIMINARY TEST RESULTS AND SUMMARY

A prototype of precision sample manipulator APS Z4-41 has been designed and constructed for the coherent surface scattering imaging project at the APS sector 8. Figure 8 is a photograph of the APS Z4-41 precision sample manipulator. The mechanical structure design of the prototype manipulator is compatible with ultra-high vacuum (UHV) requirements, although the commercial stages and actuators are high-vacuum (HV) compatible for the initial test. Table 1 summarizes the design specifications of the flexure stages for the sample manipulator system.



Figure 8. Photograph of the APS Z4-41 precision sample manipulator. The manipulator base stage PI™ Micos™ PRS-110 is not in the picture.

The Z4-4101 flexure stage's fine motion is driven by a PI 841.10V PZT actuator with strain gage sensor. The stage's coarse motion is driven by a Newport™ 8310-V closed-loop Picomotor™ linear actuator with grating encoder. If needed, an Attocube™ FPS-3010 laser interferometer system could be integrated to the stage system with an Invar reference frame [6].

We have started the preliminary testing of the prototype of the Z4-4101 compact rotary flexure stage with an Attocube™ FPS-3010 laser interferometer system. Figure 9 shows a set of the laser interferometer readout of three-up and three-down ~23 nrad steps performed by the flexure stage with PI P-841.10 PZT actuator

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AN ULTRA HIGH ACCURACY 1D LASER-BASED MICROMETER FOR DIAMETER MEASUREMENT

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INTRODUCTION

The Dimensional Metrology Group at the National Institute of Standards and Technology (NIST) has developed an ultra-high accuracy laser micrometer capable of external diameter measurements with a targeted expanded uncertainty ($k=2$) of less than 20 nanometers. This instrument is an extended development of a design developed by NIST in the 1990's and has been continuously improved since its inception. This newest design takes advantage of the lessons learned through two decades of operation, process control measurements and analysis to advance the state of the art for diameter measurements within an artifact diameter range of 0.02 to 55 millimeters. The instrument makes use of low thermal expansion materials, a stable structural design, environmental control, and remote manipulation of the artifacts to address the limiting uncertainty components contained in the previous NIST 1D micrometer uncertainty budget. A review of the instrument development and components will be presented. The anticipated operational measurement uncertainty budget for this new instrument will be outlined along with the high accuracy applications that will benefit from this effort.

LASER MICROMETER HISTORY AT NIST

The NIST Dimensional Metrology Group's long history in the development of laser-based instruments for measuring artifact diameter began in the mid-1970's. The high workload of cylindrical thread measuring wires, cylinders, and plug gauges required a measurement device with a much longer and useable scale range than was available with the classic comparators. At that time, attaching a laser to a measuring machine was a novel idea. Figure 1 shows an early attempt to use a laser scale on a contact micrometer. The instrument had design limitations, but the concept showed promise and identified many directions for future improvements.

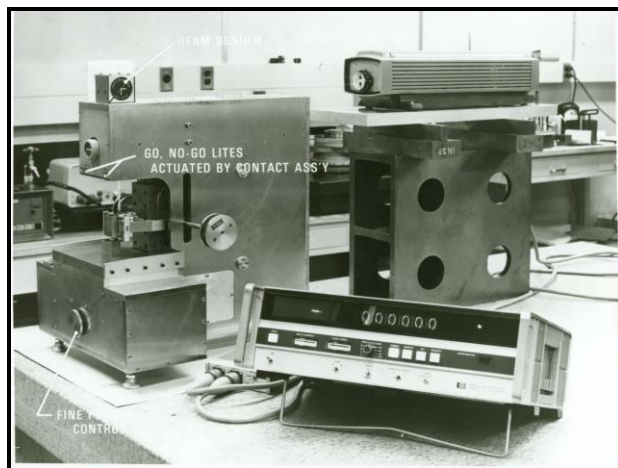


FIGURE 1. Early 1970's NIST vertical micrometer with mechanical motion and a laser scale.

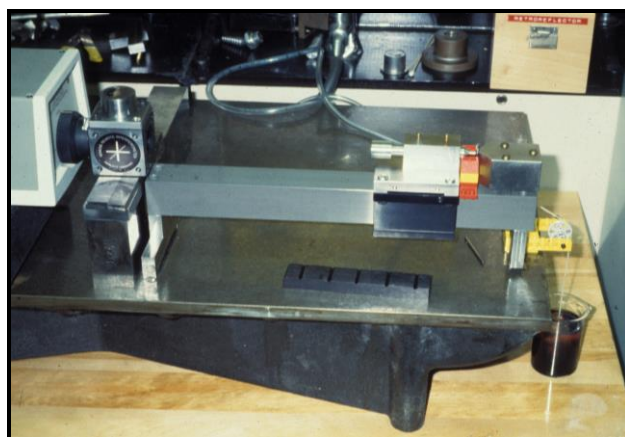


FIGURE 2. Early 1990's air bearing slide with Lego pulley and oil damper.

During the early 1990's, a renewed effort to develop a laser based micrometer began in earnest. Figure 2 shows a particularly unique approach using a Lego pulley system and a liquid

oil damper [1]. Although an amusing iteration of the concept, the importance of the test was to determine the viability of small air bearing slides to produce straight, reproducible motion. This was also the first effort to focus on horizontal displacement measurement with the retroreflector mounted on the moving carriage and the interferometer attached to the front of the bearing spine.

During the 1990's, several iterative developments led to the completed instrument shown in Figure 3. This version of the micrometer [2] has served as the basis for all comparison and absolute cylindrical diameter measurements at NIST since 2000 for artifact diameter up to 25 millimeters. Although still visually quite simple in design, several subtle design features such as fixturing techniques, locations of the optical components, contact alignment methods, material considerations, and force application techniques have produced world class measurement performance over a wide range of artifact material and sizes.

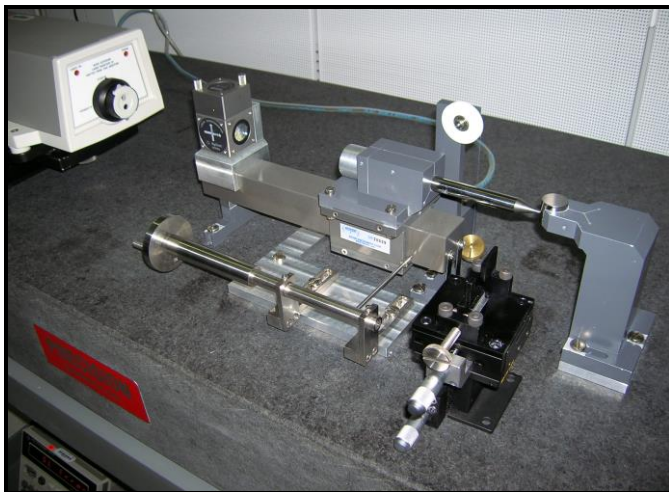


FIGURE 3. Improved development of the horizontal laser micrometer, circa 2000.

2000 MICROMETER DESIGN REVIEW

The 2000 micrometer has sufficiently met nearly all client requests for high accuracy diameter work in the past two decades. The expanded uncertainty ($k=2$) for this instrument can approach 45 nanometers for small diameter, very high quality artifacts. One of the important overlooked abilities of this instrument is its accuracy capability at the very small diameter range below 100 micrometers.

Analysis of control artifacts over the most recent 15 years of operation shows that the reproducibility performance does not deteriorate when measuring these very small fiber artifacts. The ability to vary applied measurement force and extrapolate to the zero force condition eliminates elastic deformation issues resulting from unknown material constants or the inconsistent bending behavior of the thin fibers when elastically deformed. Many of the challenging diameter requests received by NIST over the years are for the measurement of these very small, delicate artifacts.

Recently, demands for ultra-high accuracy measurements of larger diameter artifacts have also been increasing both externally and internally at NIST. Reference artifacts in the 25 millimeters to 50 millimeter diameter range for coordinate measuring machine metrology as well as requirements for deadweight pressure piston diameter measurements have pushed the current design to its practical limits, particularly for these larger sized gauges. It was clear a new instrument redesign was needed to push the expanded measurement uncertainty to below 25 nanometers for the larger sized artifacts, without reducing the effectiveness of its performance for fibers and wires less than 100 micrometers in diameter.

The new micrometer design was driven by a critical review of the 2000 micrometer uncertainty budget. This budget is heavily dependent on control artifact performance. The importance of long term control artifact data on these high accuracy absolute measuring systems cannot be overstated. It is nearly an impossible task to isolate and determine the contributions of all sources of measurement error independently. High quality control artifacts allow us to identify the larger, important components, and combined the smaller, less specific error sources together by reviewing the demonstrated reproducibility performance. Very high quality geometry control artifacts are imperative if the goal is to isolate the instrument performance from the influences of the artifact.

Several clear improvements were derived from the historical performance data and operator experience. These include a much higher quality air bearing slide coupled with a zero stress mounting method to provide more reproducible and straighter motion. Specific changes to the metrology frame material and design can provide improved

structural stiffness and beam path stability while allowing for greater flexibility and access around the artifact positioning envelope. This envelope access is critical to accommodate artifact positioning assemblies designed to be modular and flexible, allowing remote, measured movement and control of the artifact under test for better orientation and positioning accuracy.

Improved measurement uncertainty for larger diameter gauges will also require the operator to be isolated from the artifacts being measured. Combining a fixed enclosure that isolates heat from the laser source and operator from the instrument along with remote artifact positioning capability will significantly reduce the impact of environmental components in the uncertainty budget and also improve the general reproducibility of the measurement process. Gauging force application and carriage motion will still be operator controlled using manual procedures.

OVERVIEW OF THE NEW DESIGN

The structural components for the new micrometer design are shown in the schematic in Figure 4.

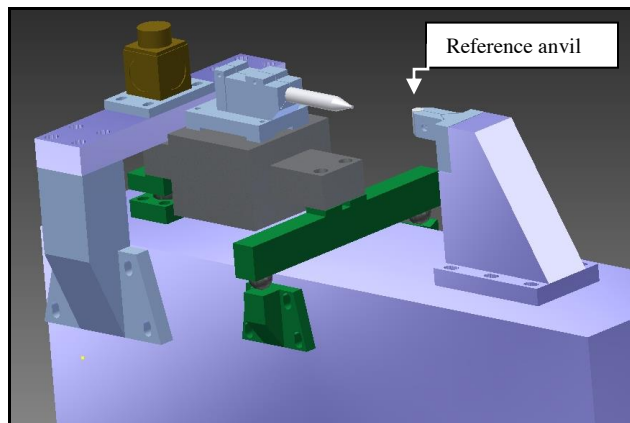


FIGURE 4. Micrometer structure redesigned.

The metrology frame of the instrument, shown in blue in Figure 4, was constructed entirely of Invar-35 material for thermal stability. The interferometer (shown in brown in Figure 4) was moved off of the air bearing to a separate bridge component that rides above and does not contact the bearing, shown in gray. The effective metrology loop, from the interferometer mount down through the heavy beam and around to the reference anvil structure, is significantly more robust and stable than earlier

designs and is completely isolated from the air bearing components. The reference anvil has been designed to be removable to accommodate different cylindrical reference sizes. This is particularly important for very small diameter fibers mounted in housing fixtures with restricted access to the fiber.

A very high quality ceramic bearing with a strong, rigid carriage design was used to supply the 1D motion. The ceramic bearing assembly is combined with multiple tungsten carbide components, shown in green in Figures 4 and 5. In addition to closely matching the thermal expansion of the ceramic bearing, these tungsten carbide components provide extra mass for the kinematic mount. The bearing assembly rests on three tungsten carbide spheres, identified in Figure 5, in a reproducible kinematic design that allows the bearing to be completely stress free during operation. The air bearing now only facilitates the 1D motion of the carriage holding the plane contact and the retro reflector.

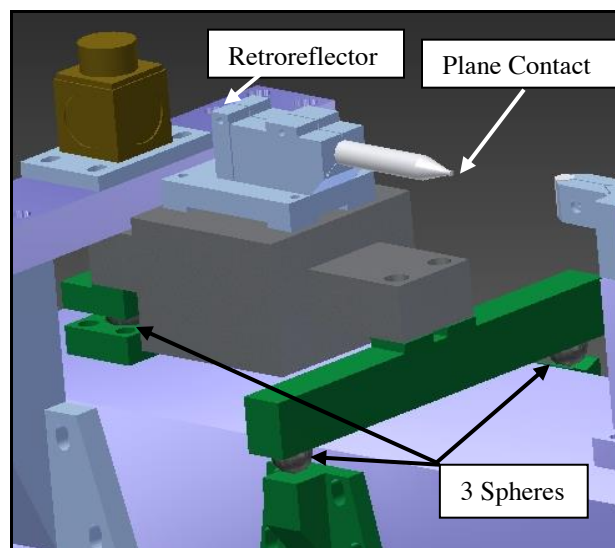


FIGURE 5. Micrometer carriage unrestricted kinematic support.

The functional range of the new micrometer was reduced from 100 millimeters to 55 millimeters. This provides a more stable optical path and results in a more compact overall design. The reference anvil and plane contacts are made of tungsten carbide to reduce elastic deformation and for control of the precision surface geometry. The hard tungsten carbide material provides reduced opportunity for damage to these important surfaces.

THE NIST ULTRA-HIGH ACCURACY MICROMETER

The completed micrometer and enclosure design is seen in Figures 6, 7, and 8. The laser source and micrometer are mounted on a mobile granite table that will allow it to be moved around the laboratory easily without disruption to the instrument alignment. The insulated foam enclosure is designed to allow full access to the instrument and the artifact mounting apparatus but still isolate the operator from the measurement. Artifact manipulation is performed from outside the enclosure using a set of precision positioning stages that slide on a fixed rail system. The artifact alignment within the micrometer contacts is accomplished using a frictionless air pad shown in Figure 7. The pad allows the cylinder artifact to self-align perpendicular to the plane contact even at low forces.



FIGURE 6. Completed micrometer with enclosure and laser placement.

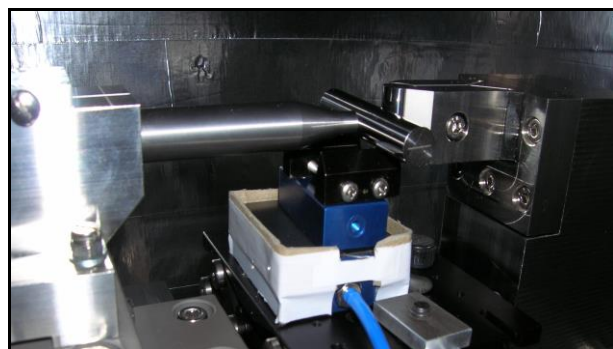


FIGURE 7. Micrometer measuring a cylinder mounted on a frictionless air pad assembly.

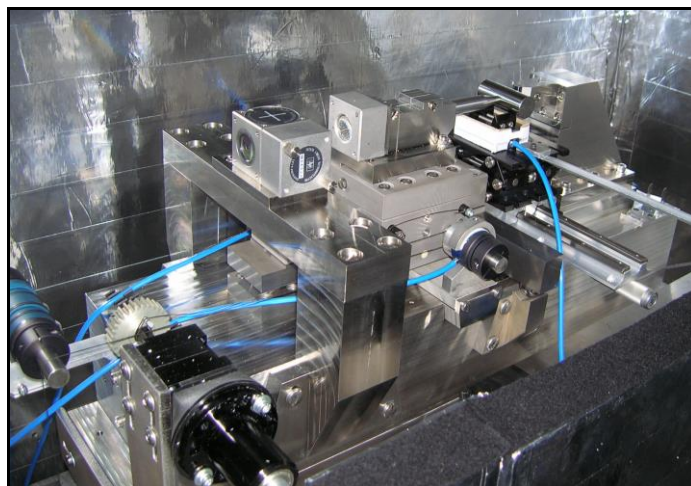


FIGURE 8. Micrometer measuring a 20.0 millimeter precision cylinder.

One of the most critical aspects of diameter measurement accuracy for the micrometer is the force application from the carriage plane contact. The instrument is designed using small frictionless air bushings and fine nylon cord to apply a constant deadweight force. These are identified in Figure 9. The force can be manipulated to any required load and is constant throughout the full measurement range of the micrometer. The retraction of the carriage is performed from outside the enclosure through a hand wheel and gear set seen in Figure 10. The gear set assembly is mounted to the heavy base of the instrument and does not affect the stability of the optical path during operation.

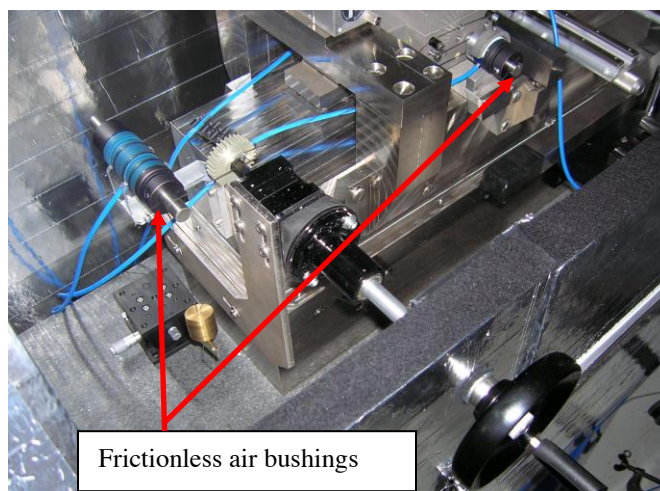


FIGURE 9. Frictionless air bushing force application. Hanging brass weights apply constant deadweight force at the plane contact.

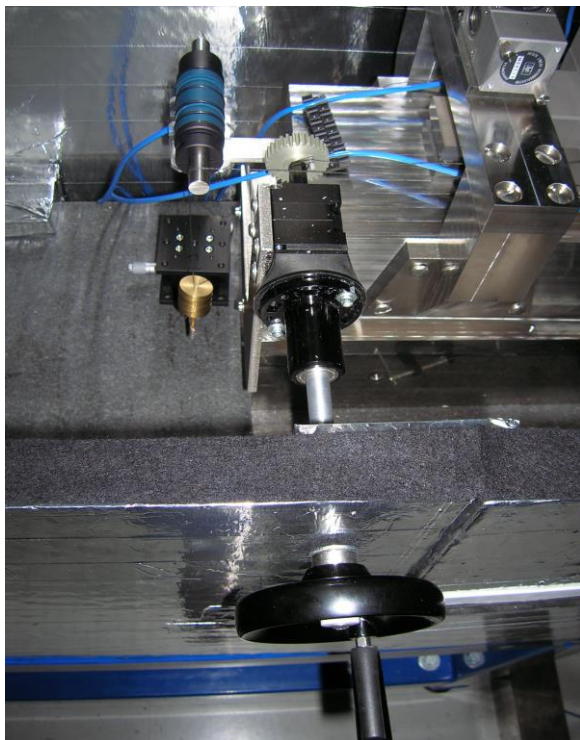


FIGURE 10. Hand wheel and gear reduction assembly for remote retraction of the carriage. The gear reduction allows for controlled, deliberate motion of the carriage for fine wire and optical fiber measurement.

UNCERTAINTY BUDGET DETAILS

The uncertainty budget for this instrument is reasonably straightforward due to its simple design and 1D performance. Current micrometer budgets indicate the major components are quantified into six primary sources.

Laser alignment and motion geometry

The laser alignment error results from laser scale misalignment to the axis of the carriage motion. For instruments with short range motion, like this micrometer, this alignment can be difficult. The high quality bearing, combined with the use of an external quadrant detector, allow us to currently align the beam to better than 3 arc-minutes over the full length of travel.

Thermal performance of the artifact

The enclosure around the micrometer will allow the thermal environment to be much better controlled and the artifact temperature more accurately

measured. Artifact temperature accuracy will be better than 4 millidegrees resulting in a standard uncertainty of about 1 nanometer on a 25 mm steel cylinder. Restricting the operating temperature range between 19.95°C to 20.05°C will dramatically reduce the uncertainty contribution resulting from the coefficient of thermal expansion uncertainty, which can be as much as 10%.

Contact Geometry

The single most critical limiting factor for measurement accuracy is the quality of the micrometer contact geometry. This term results from the imperfect geometry or alignment of the mating surfaces. This mating geometry provides the 'zero' for the instrument, and can be directly scanned and measured to locate the point(s) of definitive contact. Using tungsten contacts of optimum geometry and a plane contact length of about 3 mm, it has been demonstrated that this error can be bounded to no more than 10 nm. This would produce a standard error of about 5.8 nm.

Gauge positioning repeatability

This term results from the interplay of imperfect gauge geometry and the ability to repeatedly position and measure a specific measurand. The impact of this term is dependent on the geometric quality of the test cylinder surfaces. It is expected that with the new fixturing and positioning assembly, repeatable gauge positioning coupled with very high quality artifacts like pressure piston cylinders, this term will be reduced to 6 nm. New control artifacts will be characterized to determine if the improved positioning capability will allow this term to be sampled by the control artifact reproducibility data.

Elastic deformation correction

For the most precise effort, the test cylinder is measured at various applied forces and the results extrapolated to a zero contact force to remove uncertainty contributions from the material constants of both the cylinder and the micrometer contacts. The effectiveness of this extrapolation is limited by the range of the selected applied forces, the stability of the force application and measurement technique, and the ability to maintain a common point measurand during the data collection.

The improved ability to repeatedly reposition the artifacts coupled with lower standard applied forces

made possible by the improved bearing design and frictionless deadweight system, will result in a more repeatable force application and measurement. It is expected to lower the uncertainty of the force extrapolation to better than 4 nanometers.

Control artifact reproducibility

The measurement performance of very high quality control master cylinders of various sizes and materials is indispensable for quantifying a wide array of uncertainty components that are difficult to identify individually. Over time, these measurement histories can fully sample many small independent terms, particularly those related to environmental sensor calibration, along with those uncertainty terms that are difficult to quantify such as operator effects, temperature gradients during the measurement, residual contamination following cleaning, nonlinear thermal drift, nonlinear bending of the micrometer frame or bearing during measurement, and force measurement error. The data can be separated into length dependent and non-length dependent terms through analysis.

The new micrometer design will reduce the effects of many of these error sources through the strengthening of the frame structure, eliminating carriage friction, reducing thermal gradients from the bearing exhaust air and operator influence using remote manipulation of the gauges, and allowing better temperature measurement due to the stable thermal environment. The anticipated reproducibility improvement is hard to predict, however, we estimate that a majority of the historical effects are temperature related or related to artifact positioning repeatability. This will likely allow the artifact positioning/geometry term to be sampled by the reproducibility data as well. We estimate this reproducibility term will drop to near $0.005 + 0.20L$.

The combination of these terms in the full uncertainty budget yields an anticipated expanded uncertainty ($k=2$) of about $0.016 + 0.42L$, where L is in meters [3]. This represents a nearly threefold improvement from the previous NIST instrument, and is at state of the art levels.

DISCLAIMER

This instrument is designed for specific NMI functions and is not designed for high throughput or fully automated operation. Diameter measurements at this accuracy and uncertainty level are heavily dependent on artifact surface and geometric

quality. Client requests for ultra-high accuracy measurements carry with them a heavy burden on the client, as the usefulness of the ultra-high accuracy results demands exceptional environmental and artifact control and experienced dimensional metrology personnel to properly integrate the measurand precision into their particular project requirements.

FUTURE PLANS

Following the full development of control data on well-established precision cylinders and reference artifacts, the instrument will be fitted with a plane mirror interferometer for increased resolution and positional accuracy. We will continue to develop the flexibility of the artifact positioning assemblies to possibly allow for more than one cylinder at a time being staged for measurement, particularly small diameter wires. Elastic deformation performance will continue to be explored using a combination of different sized reference anvils and variable force application.

The M48 CMM [4] will take over all artifact measurements of diameter greater than 50 millimeters due to its premier thermal environment, fixturing, and positioning capabilities. The M48 CMM will also take advantage of new micrometer-calibrated cylindrical reference artifacts, 25 to 50 millimeters in diameter, for improved CMM probe mastering effectiveness.

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ENHANCED BIDIRECTIONAL FABRICATION OF RIGHT TRIANGULAR PRISMATIC RETROREFLECTORS

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INTRODUCTION

A retroreflector (RR) is a passive optical element whose primary function is to return the incident light back to its originating source (Figure 1). While the applications of these optical structures are widespread, the primary emphasis of this paper will be the automotive lighting applications, in which RRs are meant to enhance traffic safety by increasing the visibility of the vehicles on which they are installed to the surrounding environment/drivers.

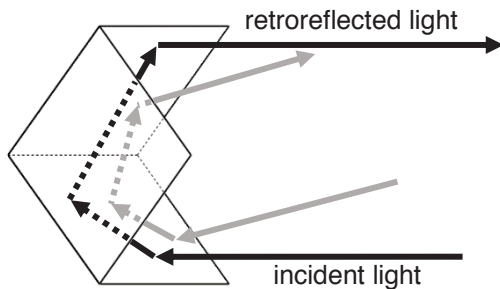


FIGURE 1. Functionality of the corner cube RR.

There are two basic classifications of retroreflective structures: lens-and-mirror, and corner cube (CC). The CC structure is characterized by a superior retroreflectivity across a wide range of incidence angles. Similar to any cubic geometry, CC is comprised of three faces that are mutually orthogonal and share a common vertex. Depending on the aperture/final configuration of the RR, the three facets of the CC can be either squared or triangular and their shapes are determinant for their retroreflective characteristics as well as their manufacturability [1]. The square sided structure has a hexagonal cross-section and higher efficiency when compared to the structure with a triangular aperture [2]. For this reason, the structure primarily used in automotive applications is the corner cube characterized by a hexagonal aperture. Fabrication of the RR arrays is a time consuming process, partly because of the sharp

90° concave corners that prevent the use the non-rotational tools.

To overcome this limitation, a different manufacturing path has been devised in the late 1970s [3]. The pin-bundling technique involves the use of hexagonal pins terminated with a single “negative” RR structure machined and then lapped at the “forming end” of the pin (Figure 2). Individual pins are bundled into an array and electroplated with Nickel forming an electroform. The electroform is then used as a cavity insert for the injection molding die. The surface finish of the RR structures must be high since an R_a of 10 nm is usually quoted as a requirement of optical surface functionality [4]. If the surface quality is low, the structure will scatter more light than desired and its optical performance will degrade.

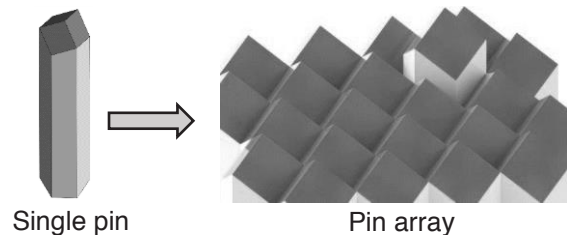


FIGURE 2. Pin-bundling technology.

Typical structure sizes used in automotive applications are about 2-3 mm and an array may have a surface area of 100 mm². Therefore, thousands of pins are commonly involved in this fabrication technique and their number is inversely proportional with the size of the RRs.

The placement of pins on non-flat or flat and inclined surfaces leads to the formation of manufacturing artifacts called “pockets” (Figure 3) whose role on the optical performance of the RRs is rather unclear at this time, even though it has been suggested that they might be associated with optical performance decreases [1].

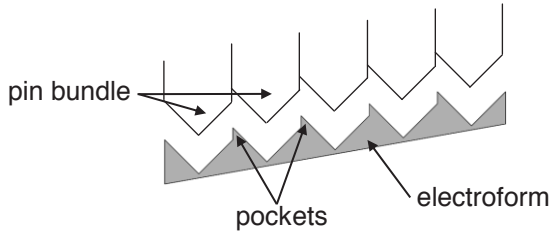


FIGURE 3. Manufacturing artifacts as a result of pin-bundling.

Extensive research efforts were recently made to reduce the overall manufacturing time associated with the pin-bundling technology that requires in excess of hundreds of hours per insert. One of the newest initiatives in this regard advocates for the use of the cutting techniques.

The switch from pin-bundling/forming to material removal/machining technologies enables a greater flexibility in terms of RR geometry and – when accompanied by optimized cutting strategies – an increased productivity. As mentioned above, the nature of the geometry necessary for RR prevents the use of classical tool path planning strategies associated with milling operations [5].

Along these lines, diamond micro-chiseling (DMC) was identified as a viable alternative to pin-bundling [2, 6, 7]. DMC has also been utilized to fabricate structure sizes as small as $150\ \mu\text{m}$. To date, DMC was solely presented in the context of hexagonal corner cube RRs (Figure 1 and Figure 2) that were cut in a flat and horizontal base surface. This technology allows for a direct machining of the injection moulding die by means of a V-shaped diamond tool whose motions are controlled by an ultraprecise five-axis micromachining system. The diamond tool cuts each RR structure in multiple layers in order to improve the final surface finish and reduce tool wear.

As alternative to DMC, ultraprecise single point inverted cutting (USPIC) was introduced as a viable option to generate RRs characterized by right triangular prismatic (RTP) geometries [6].

GEOMETRY AND OPTICAL PERFORMANCE OF RIGHT TRIANGULAR PRISMATIC RRs

Retroreflectors are generally classified according to the shape of their aperture as well as the number of facets involved in retroreflection. The right triangular prismatic

geometry is characterized by a rectangular aperture through which the incoming light rays enter and later exit, as well as two rectangular facets which reflect light through total internal reflection (Figure 4a). Unlike the CC geometry, the aperture of the RTP is not limited to a single aspect ratio, and while different geometric combinations can be imagined, the current study will be focused on a rectangular aperture determined by an isosceles aspect of the triangular facets ($\beta = 45^\circ$).

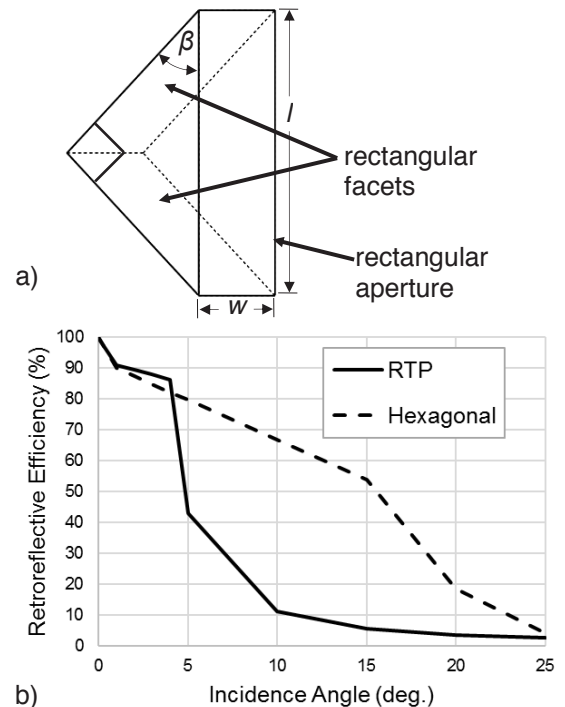


FIGURE 4. a) principal geometric parameters of an RTP, and b) retroreflective characteristics of RTP and hexagonal.

The RTP geometry analyzed in this work was rarely – if ever – part of the RR taxonomy since it was never fabricated before. Recent optical simulation studies have shown that the retroreflective efficiency of the RTP is comparable to that of the ubiquitous CC geometry that is even outperformed at incidence angles smaller than 4° [1].

DIAMOND MICRO-CHISELING TECHNIQUE

In order to achieve the required geometric accuracy and optical functionality, corner cube RRs have been produced by means of the ultraprecise five-axis CNC machining system and diamond tooling. The geometry of the diamond tool involved in DMC resembles that of

a 50° V-shaped turning tool, yet is specific to the DMC process (Figure 5). For DMC purpose, the tool was aligned in the machine such that the conventional clearance and rake faces have been switched and therefore the actual/effective cutting mechanics is different than that of turning, even if the tool is almost the same. The tool features a 22° rake angle and the clearance face has an angle of 2-3° [5].

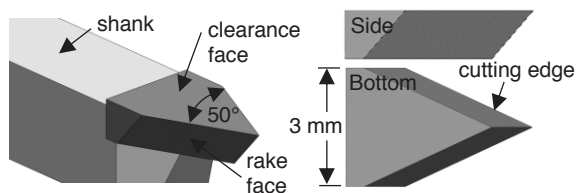


FIGURE 5. Geometry of the DMC tool.

The two rotary axes of the five-axis machining system are required for the alignment of the tool with the facets of the structure to be cut. The machine used in the DMC process had a five-axis head/table tilting or hybrid configuration in a sense that the tool can be rotated around B-axis, while the workpiece can be reoriented by rotating C-axis.

In DMC, the cutting process follows a sequence of motions in which successive layers of material are removed through cutting motions. For this purpose, the clearance face of the tool has to be perfectly aligned with the plane of the RR facet to be cut. From this position, the tip of the tool is moved linearly along the edge of the facet in a two-step process: i) plunge along the first edge to the apex, followed by ii) retract along the second edge (Figure 6). In this initial V-motion, a single facet of the CC structure is completed. The remaining two facets of the structure are completed in the same manner but only after the workpiece has been rotated 120° about C-axis in order to expose to the cutter the next RR facet.

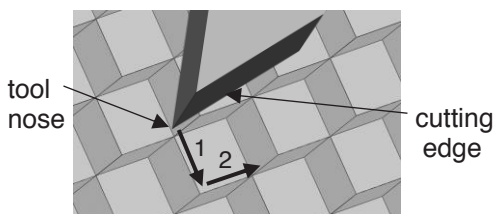


FIGURE 6. DMC kinematics.

This process of cutting and rotating is repeated for each facet of each structure in the array. It is also important to point out that after each of the indexing motions performed around C-axis, the relative position between the tool and the RR to be cut has to be adjusted by means of translational motions.

ULTRAPRECISE SINGLE POINT INVERTED CUTTING OF RIGHT TRIANGULAR PRISMS

As illustrated by Figure 7, USPIC enables the fabrication of pocketless RTPs on flat, but inclined base surfaces.

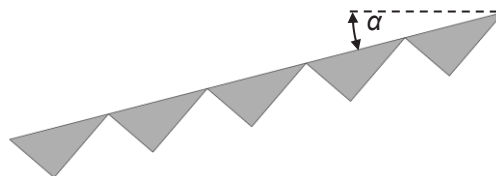


FIGURE 7. Pocketless array of RTPs fabricated on an inclined flat surface.

Similar to DMC, USPIC makes use of a monocrystalline diamond (MCD) cutting tool as mandatory prerequisite for adequate retroreflective efficiency (and hence the term “ultraprecise”). However, the geometry of the tool used in USPIC operations has been tailored to RTP fabrication and therefore it is significantly different than that used in the fabrication of CC geometries. While having a custom-design, the geometry of the USPIC cutting tool is comparable to that of a grooving tool that typically used in turning operations (Figure 8). The wedge angle was chosen to be 50° in order to prevent the adherence of the chips to the adjacent RTP facet, especially since this could potentially damage its surface finish as the tool reaches the root of each cut. This angle does contribute to a large positive rake angle that is far from being ideal for machining purposes [8], and therefore could be regarded as more of an RTP-fabrication compromise. The width of the tool determines the smallest possible structure size, but larger structures are possible by effectively cutting multiple structures next to one another. For the purpose of this study, the cutting tool width was chosen at 450 μm.

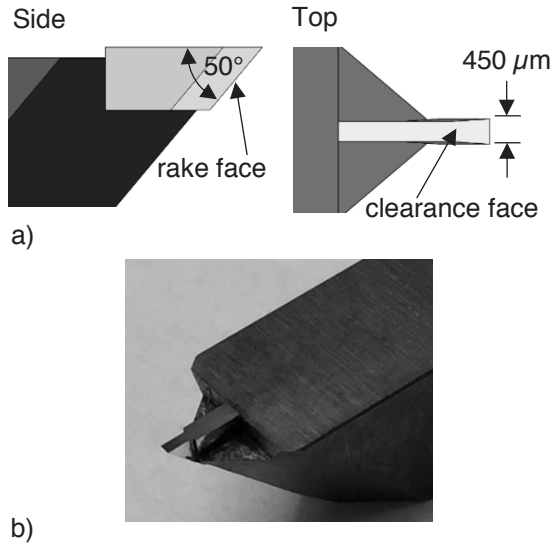


FIGURE 8. Geometry of USPIC tool: a) CAD rendering, and b) physical tool.

Unidirectional Strategy

The kinematics of this strategy requires completely different cutting mechanics for each of the two facets of the RTP structure. As depicted in Figure 9, the cutting tool engages the vertical facet in a way that is conducive for the achievement of a higher quality surface finish, while the horizontal facet is engaged in a manner that typically results in a lower surface finish. The vertical and horizontal cutting motions have been termed plunging and ploughing.

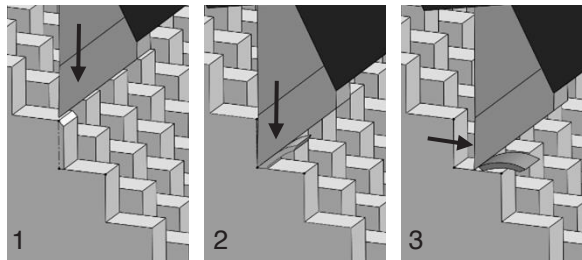


FIGURE 9. Unidirectional USPIC cutting strategy.

In this regard, plough cutting is associated with a clearance angle of at least 90°. This configuration results in a negative rake angle equal to or greater than the wedge angle. By contrast, plunge cutting is associated with a cutting mechanics characterized by a small clearance angle and positive rake angle. Since the surfaces cut through plunge or plough cutting contribute directly to the optical

functionality of the RTP, this unidirectional strategy tends to be insufficient/inappropriate to generate the required surface finish. However, its high material removal rate makes this strategy a good candidate for roughing procedures in which surface finish is not the main concern.

Bidirectional Strategy

By contrast with unidirectional approaches, the kinematics of bidirectional strategies results in superior cutting conditions for both RTP facets (Figure 10). However, while three translational motions are sufficient for unidirectional instances, bidirectional cutting scenarios typically require the full functionality of a five-axis machining system.

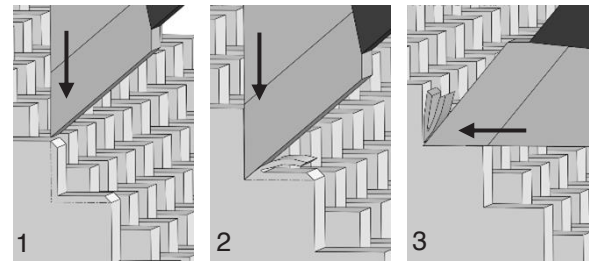


FIGURE 10. Bidirectional USPIC cutting strategy.

The primary role of the two rotary axes of the five-axis machine is to adequately align the cutting tool with the RTP facets to be cut. This type of machining is also known as 3½-axis machining, because the rotary axes are held stationary as the tool removes material [9]. In this bidirectional approach, the two active facets of the RTP are being cut alternatively (always from their “aperture-proximal end”). Because of this strategy, “ploughing” becomes unnecessary since the entire RTP can be generated by means of “plunging” motions. However, since the relatively low feed rate of the rotary axes increases the overall machining time, bidirectional approach will primarily remain an appropriate option for RTP finishing operations.

Two-Step USPIC Strategy

Following the discussion above, it becomes apparent that the best RTP-cutting strategy would encompass separate roughing and finishing procedures. Along these lines, the unidirectional USPIC strategy has been used to rough-cut the structure and resulted in a near-complete geometry, while the bidirectional

USPIC strategy has been used as a finishing procedure.

This strategy represents a further iteration of the original $3\frac{1}{2}$ -axis cutting that was introduced in the past [10]. While most of the process setting and parameters were kept unchanged, a special calibration protocol was devised and used to ensure the post-indexing parallelism between the RTP facet and the rake face of the diamond tool. In this context, it is important to note that the calibration procedure requires that the tool be within $\pm 1\ \mu\text{m}$ in order for the chips to be properly released from the workpiece throughout cutting.

EXPERIMENTAL VALIDATION

Figure 11b shows a representative sample of a planar array of RTP structures that was cut in 0.5 mm thick transparent PMMA. The array is composed of 613 structures, each characterized by the following dimensional parameters: $w = 450\ \mu\text{m}$, $l = 450\ \mu\text{m}$, and $R_a < 180\ \text{nm}$. The preliminary qualitative inspection performed on both specimens presented in Figure 11 has revealed that they are characterized by comparable optical performances.

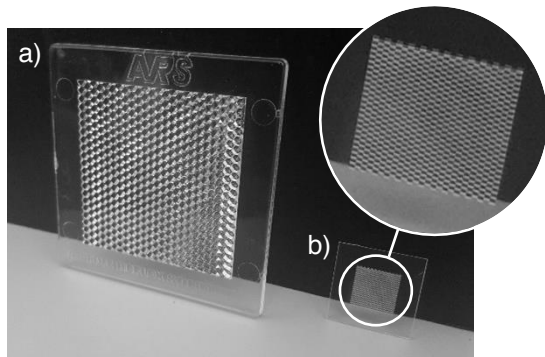


FIGURE 11. a) Sample array of traditional hexagonal RRs via pin-bundling technique b) Fabricated array of RTPs via USPIC strategy.

To further explore this aspect from a quantitative perspective, optical analysis was performed by means of a commercial luminance measurement system. The average luminance values measured for RTP and hexagonal CC arrays were $5.6\ \text{cd/m}^2$ and $10.6\ \text{cd/m}^2$ respectively (Figure 12). Even though the RTP array fabricated through the novel two-step USPIC technique was characterized by a lower optical performance, the preliminary tests have indicated that the novel strategy can generate

optically-functional retroreflective components. Without entering too many speculative – at this moment – details, it appears that the primary direction of improvement for the optical performance of the RTP RRs is tightly correlated with the overall accuracy and quality of the RTP geometry.

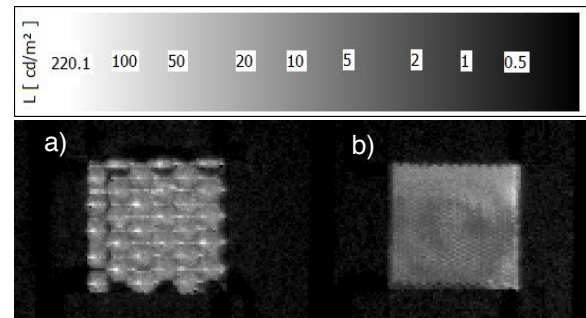


FIGURE 12. Optical Analysis using Luminance Measurement System

CONCLUSIONS

The presented enhanced USPIC technique represents a viable option for the fabrication of the RTPs. It is important to note that while USPIC was developed as an alternative for mold fabrication, its use on PMMA enables the immediate validation of the optical performance of the generated RR arrays. Future efforts will be made towards the improvement of the overall dimensional and roughness characteristics of the right triangular prismatic retroreflectors.

ACKNOWLEDGEMENTS

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HIGH SPEED MACHINING OF BRITTLE MATERIALS FOR INFRARED OPTICS

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INTRODUCTION

Ultra-precision diamond machining has been successfully producing infrared (IR) optics for many years. Diamond turning is often a cost effective method for manufacturing axis-symmetric infrared optics. However, in recent years, with the increased importance of freeform optics [1], diamond milling [2], multi-axis flycutting [3] and coordinated axis turning (slow-tool-servo (STS) and fast-tool-servo (FTS)) have developed [4,5] as production methods. However, the speed of diamond machining processes, particularly diamond milling, remain relatively uncompetitive for producing large quantities of optics. For this and other reasons, moldable infrared materials are becoming more widely used. Molding is typically more cost effective for large quantities of optics than diamond machining processes. Nonetheless, diamond machining remains one of the most viable process for production of molds for these processes and the precision of molding is material limited and is often still not sufficient to produce many optics. Therefore, it is necessary to increase the speeds of diamond machining processes both for the production of molds, for the production of prototype optics, and for the direct production of optics, particularly when tolerances are tight and the surface is freeform and has high complexity [1]. For these reasons, we seek to push the limits of coordinated axis machining and diamond ball milling process for the production of IR optics. This requires a concerted effort in: (1) understanding of the basic mechanics of machining of brittle IR materials [6] particularly in interrupted cutting processes like diamond milling; (2) machine programming and motion control and (3) software error correction. In this paper, we focus on the "high-speed" precision machining of

brittle IR materials. We discuss progress in these critical areas and indicate areas where further research is needed. We focus here on the development of high-speed cutting parameters for two very important IR materials chalcogenide glass As₄₀Se₆₀ (trade name IRG 26) and germanium as an examples. However, the same approach can be used for other materials including other chalcogenide glasses, zinc selenide, zinc sulfide, etc.

DIAMOND MACHINING PROCESSES

There are many configurations for diamond machining of brittle materials. One method of classifying them is geometry and number of degrees-of-freedom (DOF). Methods include: (1) orthogonal turning with a static tool, rotating part, simplified chip geometry, and nearly plane-strain deformations; (2) turning with a static tool, rotating part, and 2 DOF; (3) STS or FTS with a rotating part, coordinated tool motion, and 3 DOF; (4) flycutting with a rotating tool, static part, and 2 to 3 DOF; and (5) milling with a rotating tool, static part, and 3 to 5 DOF.

Orthogonal turning is a highly simplified cutting process typically used to better understand the fundamental cutting mechanics and material behavior. Intuition gained from orthogonal turning can be used to develop parameters for more complex processes. Turning can be used to create rotationally invariant surfaces including spheres and aspheres. STS and FTS can produce freeform geometries but is limited by reach and access, oscillations in cutting geometry that negatively affect cutting mechanics, total stroke and bandwidth. Flycutting is used for creating various surfaces including flats, spheres, aspheres, and freeforms but can again be limited by reach and access

and cutting mechanics. Milling utilizes 3-5 degrees of freedom to create nearly arbitrary surfaces including but not limited to those listed above. Flycutting and milling are characterized by the naturally interrupted nature of the cutting process.

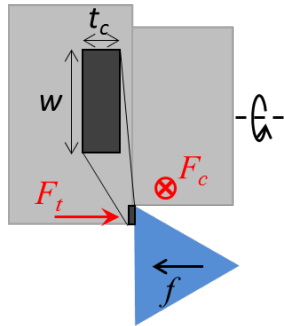


FIGURE 1. Orthogonal turning schematic

In this paper, we discuss each of these processes including physical understanding, parameter development and the cost-effective production of complex IR optics.

EXPERIMENTAL SETUP AND RESULTS

As examples of important IR materials, all experiments described here were performed in chalcogenide glass IRG 26 and germanium on a Moore Nanotechnology 350FG diamond machining center. The machine has three linear axes (X, Y, and Z), one rotational main spindle axis that could be used in positioning or turning (10 kRPM) mode (C) and one high-speed 60 kRPM milling spindle.

Orthogonal Turning

Brittle IR materials have been diamond turned for more than two decades, and there are commonalities in their behavior; we usually refer to the cutting mechanics as “ductile-dominated” or “brittle-dominated” cutting, and the goal is “ductile dominated” cutting that produces an adequate surface and subsurface quality. Orthogonal cutting is a method for gaining intuition about this behavior.

We demonstrate typical results from orthogonal cutting for chalcogenide glass IRG 26. Cutting was performed using the X and Z axes with the main spindle. Forces were measured using a Kistler 9256C2 mini-dynamometer. A 0° rake, 60° included angle, dead sharp tool positioned with the left edge parallel with the X axis was used. The tool was fed down the outer diameter of a bar of IRG 26. Surface speeds were held

constant at 4 m/s (approximately 2000 RPM). The width of the chip (w) was held constant at 200 μm . Chip thickness (t_c) was varied from 0.100 μm to 8 μm .

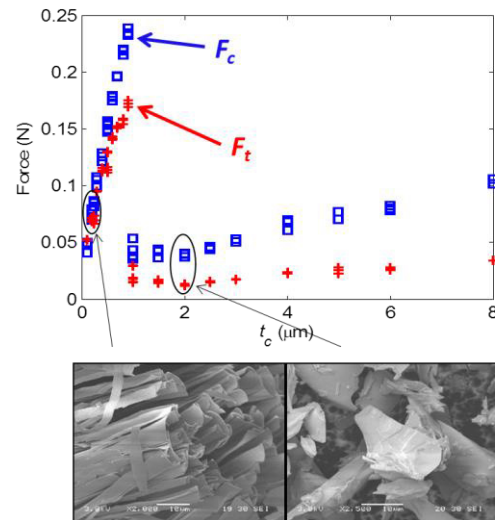


FIGURE 3. Cutting forces versus chip thickness in orthogonal cutting of chalcogenide glass and SEM images of chips

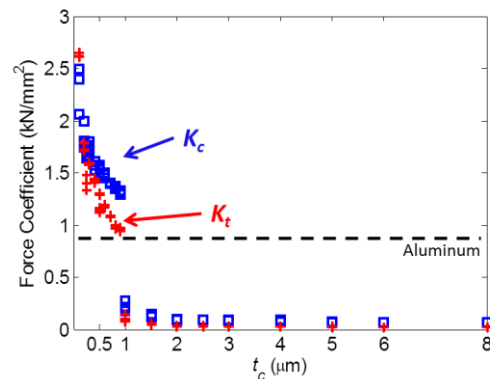


FIGURE 4. Cutting Force Coefficients from orthogonal turning of IRG26

The results and analysis from the orthogonal turning of chalcogenide glass IRG 26 can be found in FIGURE 2 and FIGURE 3. The force increases steadily as chip thickness increased until the chip thickness reached approximately 1 μm when the force decreases abruptly by approximately 5 times. SEM images of chips shown in FIGURE 2 indicate that at chip thickness less than 1 μm the chip appears to be the result of ductile deformation and at a chip thickness greater than 1 μm a fractured chip is evident.

Force coefficients show a similar drop at $1\ \mu\text{m}$ indicating that the energy required to remove material drops significantly. Prior to the drop, forces coefficients are comparable to soft metal, but after the transition the force coefficients level off at values consistent with plastics. Orthogonal cutting tests in other materials such as germanium [3,6] show similar characteristics indicating a transition in cutting mechanics at a lower chip thickness, but this transition is much more gradual than that shown here for IRG 26 glass. The differences and similarities between the behavior of the two materials is still an open research question.

Turning/Facing

The turning/facing tests shown here are for IRG 26 glass. These were performed using the X and Z axes and main spindle that was used in the orthogonal turning. In these turning tests, the feed direction was from the outer edge of the part towards the center of the spindle rotation. A $0.5\ \text{mm}$ radius single crystal diamond with a 0° rake angle was used. The depth of cut was held at $0.025\ \mu\text{m}$, the cutting speed was a constant $4\ \text{m/s}$, and the feed per revolution (fpr) was varied from $0.25\ \mu\text{m/rev}$ to $10\ \mu\text{m/rev}$. The sample was previously machined with a finish pass at $0.1\ \mu\text{m/rev}$ to minimize the effects of residual damage. Cutting and thrust forces were again measured.

FIGURE 4 Error! Reference source not found. shows cutting (blue) and thrust forces (red) as a function of fpr. SEM insets show corresponding surface characteristics. The surface at the lowest fpr showed no visible defects and had a $1\ \text{nm RMS}$ surface finish measured with scanning white light interferometry. The highest feed produced a surface with significant fracture and a greatly degraded surface finish so that it could not be quantitatively measured with SWLI due to data drop-out.

Coordinated axis turning

Results of coordinated axis turning are again shown here for IRG 26 glass. Coordinated axis turning of freeforms uses much of the current understand of diamond turning of brittle materials to determine if fracture will occur. However, one aspect that is not well understood is the effect of time-varying rake angle on the onset of fracture. When the optical surface is rotationally invariant there is no change in slope in the $\delta z/\delta\theta$ direction. However, simply adding a single freeform term like astigmatism

will create slope variation as a function of angular position of the spindle.

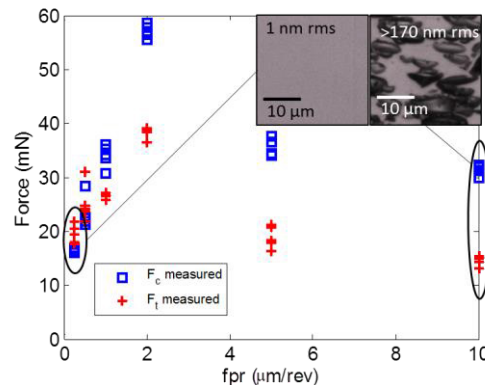


FIGURE 7. Turning force measurements with inset microscope images showing surface quality after turning

For one of the target freeforms discussed below, the prescription of the optical surface a $\pm 1.5^\circ$ degree slope variation was found in the $\delta z/\delta\theta$ direction. To understand the effect of varying slope (and therefore rake angle), a sine-wave plate was machined such that during a turning operation, effective rake angle would cycle between $\pm 3^\circ$ three times per revolution. After machining, regions corresponding to varying rakes were examined using SWLI. For this specimen, it was found that the small positive rake angles such as to be encountered during machining were not found to significantly affect cutting mechanics [7]

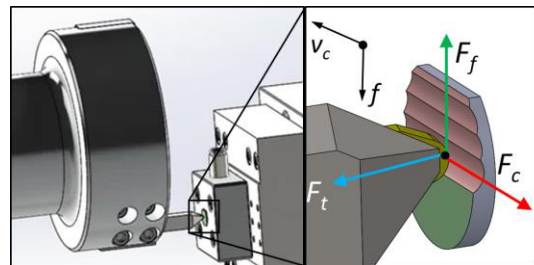


FIGURE 8. Flycutting setup showing a closer look at the tool/workpiece interaction with force directions

Flycutting

To demonstrate some of the similar aspects seen in flycutting, results are shown here for germanium. Depending on the configuration used when flycutting, freeform surfaces can be produced, however, suitable parameters must first be determined. To develop parameters for germanium, the configuration we used was similar to round nosed turning, except for the

dynamic/interrupted nature of the cutting. **Error! Reference source not found.** shows the setup used to mount the sample on the same dynamometer used in the orthogonal turning. The sample was held using a custom vacuum chuck bolted to the dynamometer. Crystal orientation of the germanium was consistent for all of the flycutting tests.

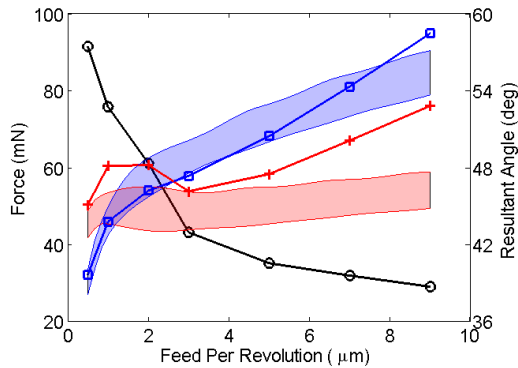


FIGURE 10. Flycutting cutting force (red) and thrust force (blue) measurements, cutting force direction angle, and force predictions (shaded).

The solid lines in FIGURE 6 show the cutting forces and resultant force angle measured experimentally. Using orthogonal cutting force data from germanium [3], the forces were also mechanistically predicted using mechanistic methods [8]. These results

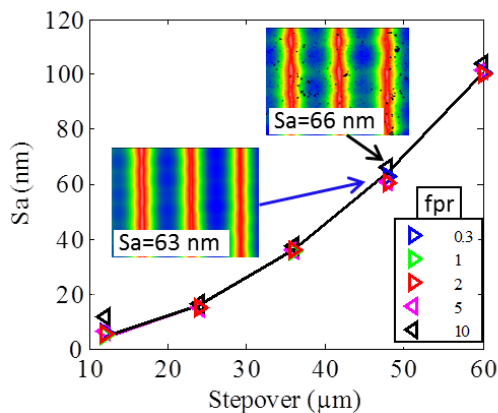


FIGURE 11. Surface finish versus stepper for ball-milled germanium force various feedrates: 0.3, 1, 2, 5, 10 $\mu\text{m}/\text{rev}$). Insets show measured surface at 50 μm stepper for lowest and highest feedrates

showed that, as with chalcogenide glass, a fracture dominated cutting occurs with increased chip thickness. The increase in chip thickness with fpr and subsequent increase in a fracture

mechanism is the cause of the change in slope of the force measurements at an fpr between 1 $\mu\text{m}/\text{rev}$ and 2 $\mu\text{m}/\text{rev}$.

Ball Milling

Ball milling is able to produce near arbitrary freeform surfaces and thus enables very complex optics. Unfortunately, it is a complex process. Ball milling generates a complex three-dimensional chip, high cutting speeds, short contact times, and dynamic tool motions due to imbalance and periodic forces. However, results of simplified cutting experiments still provide insights into the behavior of ball milling. Here we show results for ball milling of germanium.

To develop parameter sets for milling an optical surface, we conducted an array of milling tests ranging from “aggressive” roughing parameters to finishing parameters that produce acceptable surface and subsurface quality. For a spindle speed of 45 kRPM raster milling tests were conducted for stepovers ranging from 12 μm to 60 μm and fpr from 0.3 $\mu\text{m}/\text{rev}$ and 10 $\mu\text{m}/\text{rev}$. FIGURE 7 shows Sa surface finish measured with SWLI. FIGURE 7 indicates that a change in fpr has an insignificant effect on SWLI results. Yet microscopic examination shows that surface fractures begin to appear at higher fpr; the fractures are however not measured by the SWLI and so do not affect the Sa value.

$\frac{\mu\text{m}}{\text{rev}}$ $\frac{\mu\text{m}}{\mu\text{m}}$	0.3	1	2	5	10
12					
24					
36					
48					
60					

FIGURE 12. Ball mill surface finish showing the change due to stepper for 5 feed/rev (0.3, 1, 2, 5, 10 $\mu\text{m}/\text{rev}$) at each stepper

To illustrate, FIGURE 8 shows microscope images taken sequentially with equal lighting in the same instrument. The images are of each parameter patch. For the 12 μm stepper (top row) resolving the cusps is difficult. However, as the fpr and stepper are increased cusps become visible and surface pits or fractures begin to appear. The last two columns (5 $\mu\text{m}/\text{rev}$ and 10 $\mu\text{m}/\text{rev}$ feed) show increasing signs of fracture. Converting the image to gray

scale and defining a fractured section of the surface as having less than 70% maximum brightness allows qualitative identification of increased fracture. **Error! Reference source not found.** shows example histograms of microscope images with the red line indicating the 70% cutoff. At $2\text{ }\mu\text{m/rev}$ feedrate, the brightness of nearly all the pixels exceeds the threshold, whereas at a feedrate of $10\text{ }\mu\text{m/rev}$ nearly 35% of the pixels exceed the threshold.

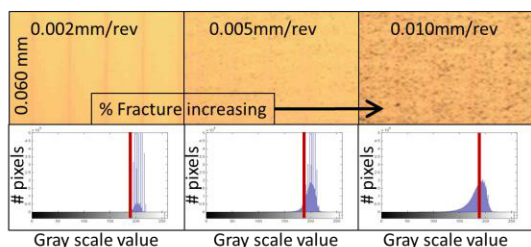


FIGURE 14. Histograms of pixel brightness for various feed rates.

FIGURE 10 shows the percentage of fracture on the surface as a function of fpr. At feed rates below $2\text{ }\mu\text{m/rev}$, virtually no fracture is indicated. At $5\text{ }\mu\text{m/rev}$ and $10\text{ }\mu\text{m/rev}$ the percentage of fractured surface increases. Fracture also tends to increase with stepover.

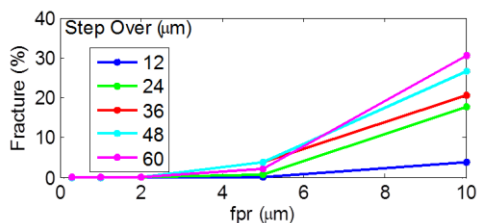


FIGURE 17. Amount of fracture determined by setting a 70% brightness threshold (Red line) on a histogram

The goal of these tests is to increase the material removal rate to the highest possible level to enable cost effective production. We now discuss applications of freeform machining to optics.

RESULTS AND APPLICATIONS

Landscape Lens

The first application (FIGURE 11) is an IRG 26 landscape lens design with two aspheric surfaces (designed by David Schmidt from Rochester Precision Optics). Based on research results discussed in the previous section, the material removal rate was $16\text{ mm}^3/\text{min}$ for roughing and $0.06\text{ mm}^3/\text{min}$ for finishing [2]. The entire optic was a freeform surface containing

the lens and kinematic grooves to demonstrate rapid assembly and alignment.

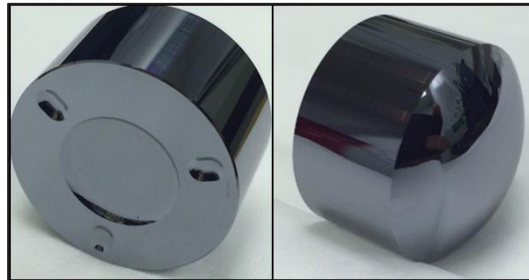


FIGURE 18. Ball milled landscape lens with kinematic mount features milled in the same operation as the concave side of the lens

Alvarez Lens

The next IRG26 optic we made was a high order Alvarez lens (design by Jason Shultz at UNC Charlotte). This surface had a “true” freeform shape with a Cartesian coordinate system equation and no axis of symmetry. This optic is described in detail by Barnhardt et al. [7].

Twisted Pyramid

To demonstrate high speed machining capability in IRG 26, a “twisted pyramid” was designed and machined using more aggressive parameters than had been previously used. The pyramid had slopes as high as 90 degrees. The roughing parameters were a 1 mm axial depth of cut, a 0.25 mm stepover with a 0.010 mm/rev feed at 40,000 RPM spindle speed: a material removal rate of $100\text{ mm}^3/\text{min}$. At these speeds, form errors due to machine axis motions become a limiting factor in the machining time.

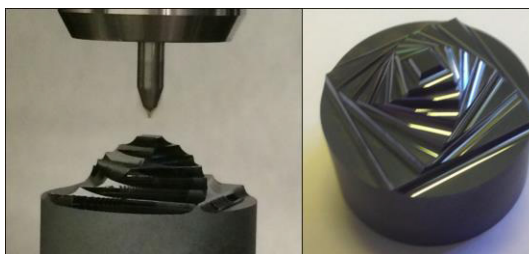


FIGURE 20. Ball milling of a severe freeform surface in IRG26.

DISCUSSION

Freeform optics are a disruptive technology in the optics industry. Diamond machining is an enabling technology particularly for IR optics where form tolerances on the order of $\pm 100\text{ nm}$ and surface finish less than 5 nm RMS are typical. However, to realize the potential, manufacturing processes need to be further developed so that it is possible to cost effectively

manufacture freeform optics at production rates. This requires future research in: (1) cutting mechanics of brittle IR materials; (2) machine motions; (3) error mitigation; (4) tooling and (5) metrology. Machining speeds particularly for versatile processes such as freeform ball milling must be increased by an order of magnitude for cost effective manufacturing by machining to become a reality.

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A COMPACT SPECTROSCOPIC SYSTEM BASED ON CUSTOM-PRINTED CONCAVE GRATINGS

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In this paper, we present the design and characterization of a compact spectrometer. The key component in the spectrometer is a concave blazed grating of varying line-space, custom-printed via our precision vacuum-imprinting system [1]. The optical quality of the concave grating was characterized, achieving a total diffraction efficiency of 85%. The compact spectrometer has a full width at half maximum (FWHM) resolution of ≤ 2.0 nm over the visible spectrum (400-700 nm) and a footprint of 90 x 88 mm², comparable with state-of-the-art commercial spectrometers.

DESIGN OF A COMPACT SPECTROGRAPH

Compact spectrometers have become increasingly important due to their portability and low cost. Accordingly, they are widely used in chemical analyses, remote sensing of chemicals, and biological research [2, 3]. To miniaturize a spectrometric system, one needs to integrate different optical components into a compact envelope; the most effective way to realize this is to combine refractive and diffractive optical elements into a new optical element, i.e., a concave grating. Figure 1 compares the concave-grating design with the Czerny-Turner design that employs a planar grating, which calls for additional space and alignment precision.

Conventionally, due to the limited manufacturing capability, e.g., holographic recording or mechanical ruling, only concave holographic gratings and convex blazed gratings can be custom-built and widely used in the compact spectrometers [4, 5]. However, concave holographic gratings have low diffraction efficiencies and convex blazed gratings lack the focusing capability, thereby limiting the performance of the spectrometers. A

customizable concave blazed grating that can simultaneously address spectral resolution and diffraction efficiency has yet to be developed. The ability to custom-design concave gratings is critical as commercially available concave blazed gratings have limited range of selections, e.g. curvature of substrates and density of lines, and cannot meet the requirements of various optical designs.

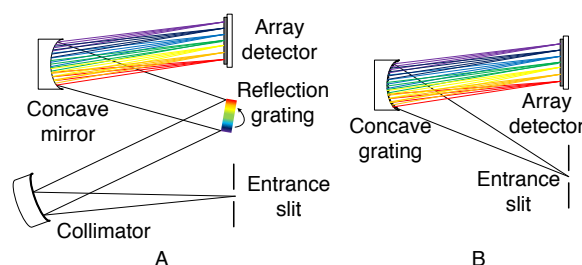


FIGURE 1. Optical design of typical compact spectrometers using a (A) planar grating, i.e., Czerny-Turner design; and (B) concave grating

In addition to customizable curvatures, to optimize the design presented in Figure 1B, a concave grating of variable line-space (VLS) should be used. As shown in Figure 2A, when a concave grating of constant pitch (or equidistant grooves) is used, i.e., Rowland spectrometer [6], the diffracted spectrum will focus on a curved plane, i.e., Rowland circle, resulting in low resolution when pairing with a flat photodetector. Contrarily, as shown in Figure 2B, a properly designed VLS concave grating can project the spectrum to a flat imaging plane, which improves the spectral resolution and minimizes astigmatism. In this work, VLS concave gratings are fabricated by exploiting the non-uniform expansion process of the elastomeric stamp during the vacuum printing process, discussed in the following sections.

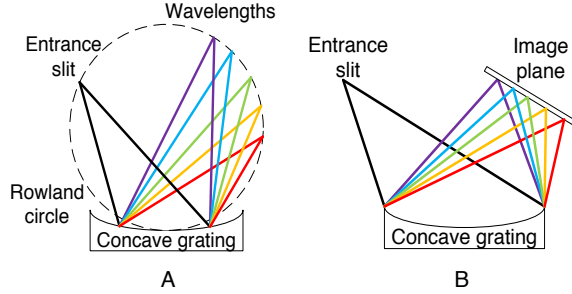


FIGURE 2. Focusing effect of a constant pitch grating (A) and a variable line-space grating (B)

DESIGN REQUIREMENTS

Design requirements are carefully generated with an aim to develop a high resolution (< 2.5 nm) compact spectrometer operated in the visible range (400 nm to 700 nm). The overall footprint of the device should be less than $10 \times 10 \times 5$ cm³. Figure 3 presents the detailed design of the compact spectrometer. The design parameters of the system are summarized in Table 1, where (r_A, θ_A) is the polar coordinate of the incident slit position; (r_{B1}, θ_{B1}) and (r_{B2}, θ_{B2}) are the detector's position; R is the radius of curvature of the concave substrate; l is the line density of the grating; and W is the slit width.

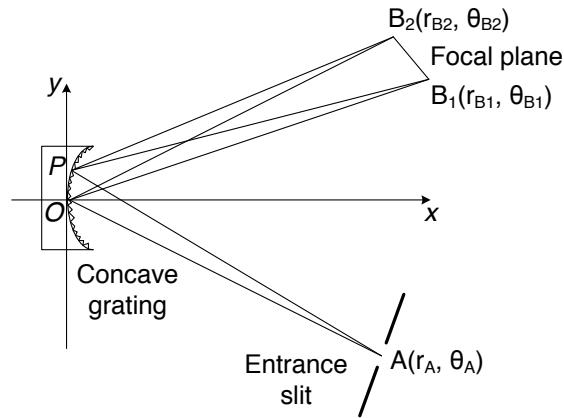


FIGURE 3. Detailed design of the compact spectrometer integrated with a custom-built VLS concave grating

TABLE 1. Design parameters

Concave grating	R	103.4 mm
	l	530 l/mm
Incident slit	W	10 μ m
Position of the slit	r_A	84.0 mm
	θ_A	20.0 °
Position of the CCD array	r_{B1}	96.1 mm
	θ_{B1}	40.6 °
	r_{B2}	96.1 mm
	θ_{B2}	48.8 °

FABRICATION OF CONCAVE BLAZED GRATING

In this section, we briefly describe the fabrication processes of the concave blazed grating. First, a concave lens of proper size and power is selected and spin-coated with UV epoxy. Next, blazed grating patterns are imprinted onto the concave lens using a custom-built vacuum nanoimprinter [1, 7], where a flexible polydimethylsiloxane (PDMS) stamp carrying grating patterns is inflated uniformly to perform pattern transfer. After UV light exposure, the blazed grating patterns on the concave lens are formed. The accuracy of the blazed grating is determined by the precise control of the (1) stamp expansion ratio, K , (2) imprint pressure, p , and (3) gap distance, d , between the stamp and the concave lens. Adjustment of the gap distance may slightly increase or decrease the grating pitch and blazed angle. The stamp expansion ratio, which is a function of the radial position and thickness of the membrane, determines the local line density of the printed grating. Experiments were devised and performed to investigate the relationship between imprinting quality and processing parameters including K , p , and d . The results are summarized in Table 2. From the study, we used $p = 0.5$ psi and $d = 25$ mm to print the concave grating for the spectrometer. After the imprint, the grating was coated with 500 nm aluminum via e-beam evaporation.

TABLE 2. Grating pitch vs. imprint pressure and gap distance

		Grating pitch	Extension ratio (K)
Imprint pressure (p)	0.5 psi	1888 nm	1.13
	1.5 psi	1928 nm	1.16
	2.5 psi	1998 nm	1.20
	3.5 psi	2102 nm	1.26
Gap distance (d)	7 mm	1725 nm	1.03
	17 mm	1998 nm	1.20
	25 mm	2141 nm	1.28

Figure 4A presents optical and AFM characterization results of the imprinted concave grating. The data confirm the sawtooth-profile of the blazed grating has been successfully transferred from the PDMS stamp with a pitch of 1888 nm, promising high grating efficiencies. Figure 4B presents the diffraction test; the

results are summarized in Table 3. The experiment was performed with a continuous wave laser (632.8 nm; 15.6 mW). From the results, we find the total efficiency reaches ~85% and the maximum intensity occurs at 632 nm at the -1st order.

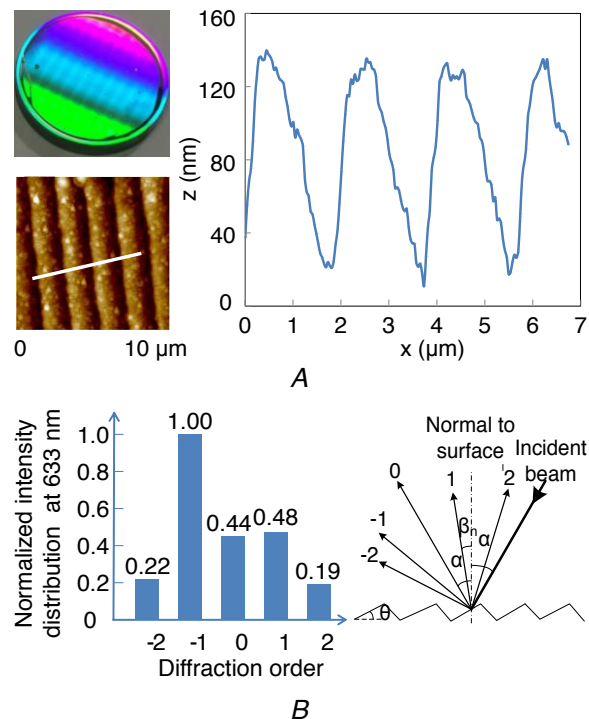


FIGURE 4. (A) Optical and AFM characterization results of the UV imprinted concave blazed grating (diameter: 25.4 mm, focal length: -200 mm); (B) diffraction test of the imprinted concave blazed grating coated with aluminum

TABLE 3. Summary of diffraction test

Order	Power (mW)	Efficiency (%)
-2	1.25	8.0
-1	5.67	36.4
0	2.52	16.2
+1	2.75	17.6
+2	1.07	6.9
Total	13.26	85.0

Next, we characterize the pitch variation of the imprinted concave grating along its radial direction from the center of the grating; the results are presented in Table 4. From the results, we find the line density of the VLS concave grating to be high in the center region and gradually decrease towards the edge of the grating, which helps to achieve a semi-flat imaging plane (versus a constant pitch grating), as shown in Figure 2B. More adjustments of the

local linewidths can further flatten the imaging plane, which may be achieved by controlling the stamp thickness and fine-tuning the imprinting parameters (p and d).

TABLE 4. VLS concave grating pitch variation along the radial direction

Radial position (mm)	Pitch (nm)
0.0	1888
2.5	2000
5.0	2113
7.5	2226
10.0	2338
12.5	2451

CHARACTERIZATION OF SPECTROMETER

In this section, we characterize the compact spectrometer. The experimental setup is shown in Figure 5, where an entrance slit of 10 μm size and a rectangular photodetector of 2048 pixels (Sony ILX511) are used. The incident beam is first projected to the concave grating with a central pitch and radius of curvature of 1888 nm and -103.4 mm respectively; the beam is then diffracted and dispersed to the linear photodetector array for spectral imaging. Three continuous wave (cw) lasers of different wavelengths (473 nm, 532 nm, and 633 nm) and small bandwidths are used to characterize the spectral resolution. The results are shown in Figure 6, where FWHM bandwidth is used to define the spectral resolution. Table 5 presents the measured spectral resolution at each wavelength.

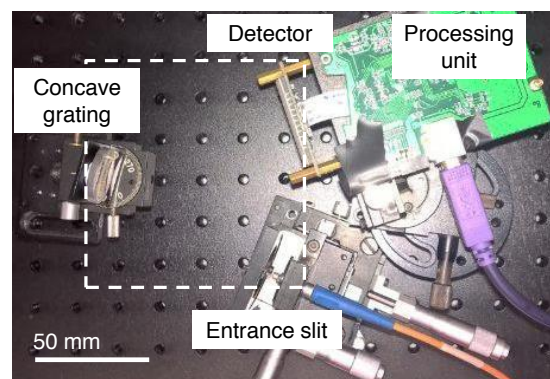


FIGURE 5. Experimental setup for characterizing spectral resolution

The overall footprint of spectrometer is determined by the location of entrance slit, the concave grating, and the photodetector shown in the box (white dashed line) in Figure 5. After optimization, the final footprint of the entire

system is $9 \times 9 \times 3 \text{ cm}^3$ with a weight of ~ 230 grams, satisfying our design goal. Table 6 compares our compact spectrometer with state-of-the-art miniature spectrometer, i.e., Torus Series, Ocean Optics. From Table 6, we find that the compact spectrometer has comparable resolution with the Torus system with a substantially reduced footprint, which improves the portability. The spectral resolution of our system may be improved by introducing a focusing lens before the linear photodetector at the expense of increasing the system footprint or by further optimizing the custom-printed VLS grating. It is worth to note that a smaller slit size may improve the spectral resolution by giving away signal strength and increasing the photodetector integration time. Given that the Torus system uses the same photodetector (Sony ILX511), our compact spectrometer may have better signal to noise ratio and faster response time.

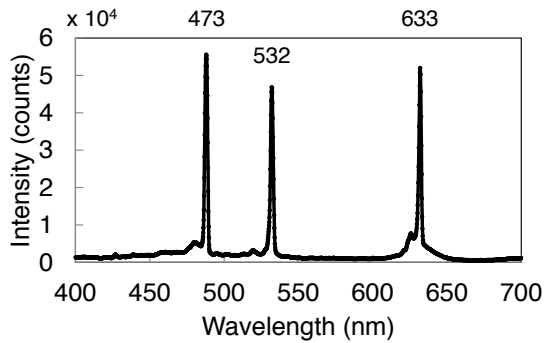


FIGURE 6. Measured spectrum of three cw lasers: diode laser (473 nm); Nd:YAG laser (532 nm); and He-Ne laser (633 nm)

TABLE 5. Spectral resolution of the compact spectrometer at different wavelengths

Wavelength (nm)	Experiment value (nm)
473	2.3
532	2.1
633	2.0

TABLE 6. Performance and footprint comparison between the compact spectrometer and Torus, Ocean Optics

Spectrometer	Compact spectrometer	Ocean Optics
Footprint (cm^3)	$9 \times 9 \times 3$	$15 \times 12 \times 6$
Resolution (nm)	2.0	1.6
Slit (μm)	10	5
Weight (g)	230	954

CONCLUSION

We present the design of a compact spectrometer with a spectral resolution of 2.0 nm and a footprint of $9 \times 9 \times 3 \text{ cm}^3$ using only one custom-built optical element. The compact design is achieved by the use of the custom-built VLS concave grating that has a diffraction efficiency of 85%, comparable to commercial gratings. The varying line-space in the grating is achieved via the precision vacuum-imprinting method.

ACKNOWLEDGMENT

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Estimation of Machine Tool Precision based on Inspection Data

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INTRODUCTION

The highly competitive markets of today's world keep pushing companies to make parts faster and with tighter tolerances than before. Traditionally to determine if a machine tool can make a part with the requested tolerance, the manufacturing engineer faces two alternatives. First is to base the decision on his experience. Second to machine a trial part, this is not the most efficient use of time and resources. A potential solution to this issue may lay in creating a full machine error budget and identifying all the error sources and their magnitudes. That solution requires a substantial time and effort. The alternative approach to estimation of manufacturing precision may be found in use of test data collected during the machine tool final inspection.

Modeling of machine tools and error budgeting has been extensively researched [1,2,3,8]. Most error budget approaches are based on the Form Shaping Function (FSF) which uses a mathematical model which represents relative displacement of the major machine element using Homogeneous Transformation Matrices (HTM).

Machined part errors on a turning center can be traced to four main sources, spindle axis motion, slide kinematics, static and dynamic compliance and thermal errors [6]. In this particular study thermal errors will not be considered since the part cycle time is relatively short (1-5min). In addition is common practice to warm up the machine before starting the production cycle.

The proposed methodology requires a FSF model of the machine tool, developed at design stage, and the measurement of the spindle axis motion and the axis straightness, obtained during machine tool final inspection in the working envelope [8].

The FSF model will compute the virtual part as a point cloud. The cylindricity, straightness, and roundness errors of this point cloud will be estimated the same way the metrology equipment measures the machined part. Proposed methodology is illustrated by comparison results of virtual and real turning of a cylinder and a sphere at turning center.

DYNAMIC MODEL OF MACHINE TOOL STRUCTURES

The part accuracy computation requires a machine tool model that allows direct computing of part profile errors, based on the machine tool layout and the machine tool structure components characteristics (kinematics errors, statics and dynamic compliance).

The kinematic models for machine tool error evaluation based on the Form Shaping System (FSS) are presented in [1,2]. The stiffness model was developed in [8] from the kinematic model [1], by adding the stiffness characteristics of the machine components. All the other required information (distances between components, motion simulation parameters, etc.) has already been defined in the kinematic model. This section presents the development of the dynamic model for a machine tool, using the same principle, which uses the stiffness model and extends it by adding simplified mass properties of the machine tool components.

At this stage the dynamic characteristics of the structure that need to be considered are the natural frequencies and frequency response function in the cutting zone.

The relationship between errors $d\vec{r}_0$ in the cutting zone and the displacement between model links $d\vec{R}_i$ are demonstrated in [8] and can be presented as:

$$d\vec{r}_0 = \sum_{i=1}^l G_i d\vec{R}_i \quad (1)$$

Where: - $d\vec{r}_0 = (dx_0, dy_0, dz_0)^T$ - differential of radius vector $\vec{r}_0$ of cutting point; small displacements (dx_i, dy_i, dz_i) and small rotations $(d\alpha_i, d\beta_i, d\gamma_i)$ of system S_i relative to system S_{i-1} are presented by vector $d\vec{R}_i = (dx_i, dy_i, dz_i, d\alpha_i, d\beta_i, d\gamma_i)^T$; G_i - (3x6) transmission matrix referring (reducing) displacement in the i-th joint to the displacement at the cutting point [8].

Vector $d\vec{R}_i = (dx_i, dy_i, dz_i, d\alpha_i, d\beta_i, d\gamma_i)^T$ is defined by a (6x1) vector of acting load $d\vec{F}_i$ at the i-th link and a (6x6) compliance matrix W_i of this joint:

$$d\vec{R}_i = W_i d\vec{F}_i \quad (2)$$

The compliance matrix in this equation is not defined explicitly and may be of static or dynamic nature. The dynamics of the rigid body with 6-DOF (without damping) are described by the Newton–Euler equation [10]:

$$(-\omega^2 M_i + K_i) d\vec{R}_i = d\vec{F}_i \quad (3)$$

Where M_i is the (6 × 6) inertia matrix of i-th link; K_i is the (6 × 6) Cartesian stiffness matrix of i-th joint; and ω is the frequency (in rad/s). It follows from comparison Eq. (2) and Eq. (3) that

$$d\vec{R}_i = (-\omega^2 M_i + K_i)^{-1} d\vec{F}_i \quad (4)$$

After substituting Eq.(4) in Eq.(1) we will get equation describing dynamic components of error in cutting zone:

$$d\vec{r}_0 = \sum_{i=1}^l G_i d\vec{R}_i = \sum_{i=1}^l G_i (-\omega^2 M_i + K_i)^{-1} d\vec{F}_i \quad (5)$$

Let's consider the case when only dynamic forces acting on the machine tool structure is the variation of the dynamic cutting force $d\vec{P}_l$:

$$d\vec{P}_l = dP\vec{\rho} = dP(\rho_x, \rho_y, \rho_z)^T \quad (6)$$

Where: $\vec{\rho}$ -unit vector of applied force, that defines its direction, and dP defines its variation.

In the simple case when variation in the cutting force is defined by variation in depth of cut h , $d\vec{P}_l$ as presented in [9] may be expressed as;

$$d\vec{P}_l = Kdh\vec{\rho} = K_p b \cdot dh \cdot \vec{\rho} = K_p b \cdot (d\vec{r} \cdot \vec{n}) \cdot \vec{\rho} = K_p b [\vec{n}^T d\vec{r}] \vec{\rho} = K_p b \cdot [\vec{\rho} \cdot \vec{n}^T] d\vec{r} \quad (7)$$

Where: $W_c = K_p b \cdot [\vec{\rho} \cdot \vec{n}^T]$ -is the (3x3) matrix of the cutting process characteristic (static or dynamic).

- $K = K_p b$ - where K_p is specific cutting force [N/m²], and b [m] – is the width of cut;

- $\vec{n} = (n_x, n_y, n_z)^T$ - unit vector normal to the cutting surface;

- $[\vec{\rho} \cdot \vec{n}^T]$ - matrix multiplication of

$\vec{\rho}$ dimension (3x1) and matrix $\vec{n}^T$ dimension (1x3).

The variation in $d\vec{r}$ may be defined as sum of two sources:

- Displacement between the cutting point and the workpiece $d\vec{r}_0$;
- Independent of motion between tool and workpiece change of depth of cut, defined, by example, by part profile - $d\vec{f}$.

Now the variation in the cutting force may be presented as follows

$$d\vec{P}_l = W_c (d\vec{r}_0 + d\vec{f}) \quad (8)$$

Variation of the cutting load $d\vec{P}_l$ action on i-th link (rigid body) may be written as:

$$d\vec{F}_i = d\vec{P}_l = G_i^T \cdot d\vec{P}_l = G_i^T W_c (d\vec{r}_0 + d\vec{f}) \quad (9)$$

And Eq. (4) rewritten as

$$d\vec{R}_i = (-\omega^2 M_i + K_i)^{-1} G_i^T W_c (d\vec{r}_0 + d\vec{f}) \quad (10)$$

After substituting $d\vec{R}_i$ from (10) in (1) we will get:

$$d\vec{r}_0 = \sum_{i=1}^l G_i d\vec{R}_i = \sum_{i=1}^l G_i (-\omega^2 M_i + K_i)^{-1} G_i^T W_c d\vec{r} \quad (11)$$

Let's define the (3x3) matrix of dynamic response function of machine tool structure in cutting zone as:

$$W_{Dyn} = \sum_{i=1}^l G_i (-\omega^2 M_i + K_i)^{-1} G_i^T \quad (12)$$

Expression (12) may be used to find the natural frequencies of the machine tool structural system by solving the eigenvalue problem:

$$\det[\sum_{i=1}^l G_i (-\omega^2 M_i + K_i)^{-1} G_i^T] = 0 \quad (13)$$

Taking into account the definition from Eq. (12) the displacement in the cutting zone Eq. (11) may be written as:

$$d\vec{r}_0 = W_{Dyn} W_c d\vec{r}_0 + W_{Dyn} W_c d\vec{f} \quad (14)$$

It follows from Eq. (13) that

$$d\vec{r}_0 = (E - W_{Dyn} W_c)^{-1} W_{Dyn} W_c d\vec{f} \quad (15)$$

Eq. (15) can be used to compute the effect of the machine tool structure dynamic and cutting process on the machine accuracy during precision cutting operations or to find the border line of stability (chatter condition) of a machine tool system as function of cutting conditions (in our case the width of cut b).

PART ERROR ESTIMATION THROUGH MODELING

Developed kinematics [1], stiffness [8], and dynamic models of machine tool may be used to estimate accuracy, productivity, and reliability of machine tool at different stages of machine tool design, manufacturing, and application.

At the application stage these models may be used to compute productivity limitations such as chatter or the coefficient of restitution. From the accuracy point of view, the stiffness model may be used to find conditions when the cutting forces are detrimental to accuracy and need to be taken in account. The dynamic model may be used to ensure that cutting conditions, for example RPM, does not coincide with the machine structure natural frequencies, etc.

In the simple case of a finishing cut the compliance errors can be neglected since the cutting force fluctuation is very small. In this case estimation of part accuracy requires only the kinematic model of the machine tool structure (developed at the design stage), the measurement of spindle axis motion, and axis straightness (both obtained during the machine tool final inspection).

EXPERIMENTAL SETUP

The spindle axis motion was measured with a LION Precision© spindle analyzer, which uses capacitance probes and a master spherical target. The Z axis straightness was measured with the LION spindle analyzer also, however instead of the spherical master, a straight edge

is used. Both experimental sets of data were used as an input to the machine tool model [8] to simulate the cutting of a virtual cylinder.

The brass cylinder was machined on the same turning center where the measurements were performed. The accuracy of the machined cylinder was measured using a Zeiss© Rondcom 54 Form Tester. The straightness of the part is measured along cylinder side on two positions 180 degrees apart. To ensure that the measured machine straightness may be used for analyses comparison of measured profile and machined part profile was performed as shown in Figure 1.

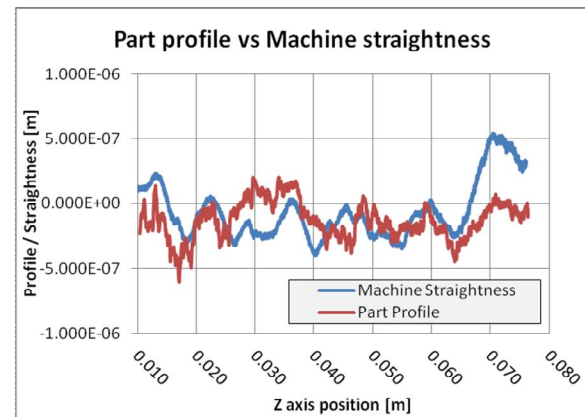


FIGURE 1. Comparison between machine straightness measurement and profile of machined part.

RESULTS

Using the spindle axis motion and straightness errors measured on the machine a virtual cutting process was simulated. A point cloud is generated, as shown in Figure 2.

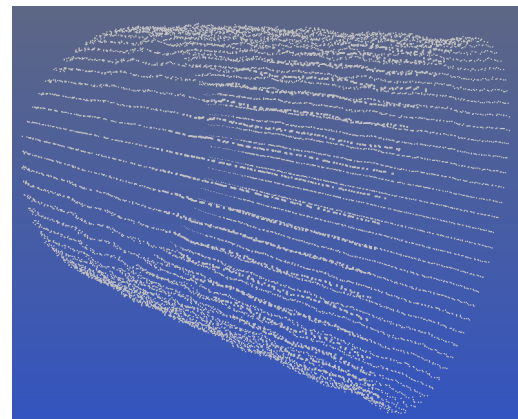


FIGURE 2. Point cloud generated during the virtual machining with the spindle axis motion and straightness errors magnified 2000 times.

A straight profile section and cross section is extracted from the point cloud to compare the straightness and the roundness of the part. The measured part roundness was $0.669\text{ }\mu\text{m}$ and the virtual part roundness was $0.656\text{ }\mu\text{m}$, see Figure 3.

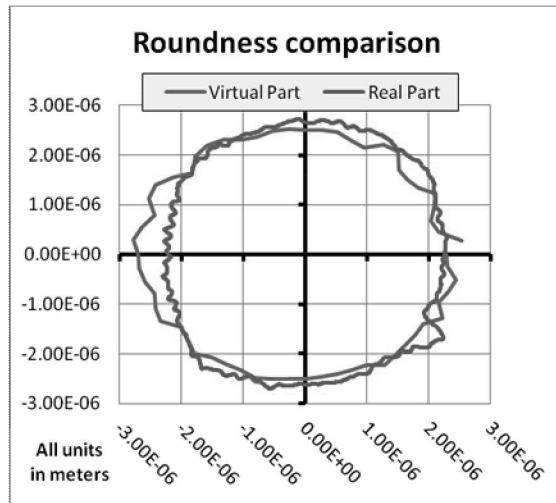


FIGURE 3. Comparison of virtual part roundness ($0.656\text{ }\mu\text{m}$) and actual part measurement ($0.669\text{ }\mu\text{m}$).

The straightness comparison between the virtual profile and the cut part profile is shown in Figure 4. As it follows from presented data the virtual part accuracy agrees with the actual machined part measurements.

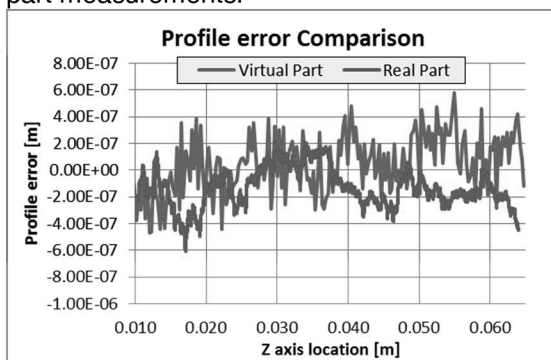


FIGURE 4. Comparison of the virtual part profile and the actual part measurement.

CONCLUSION

The proposed methodology can be used by manufacturing to check if chosen machine tool will provide the required accuracy. By using a machine model different machines and part combinations can be efficiently tested without the need of setting up work holding, tooling and material on a real machine. This task may be

performed by a FSF model of the machine tool, that uses the input data obtained by machine tool manufacturer during final inspection. For example, the motion axis straightness and the spindle axis motion.

Collecting straightness data is common for most of the machine tool builders since is part of the ISO and ANSI inspection documents. The spindle axis motion measurement is included in ASME Standard but not required and may be done in final inspection upon agreement between machine tool manufacturer and buyer.

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ELASTIC AVERAGING FEATURES FOR RAPID PRECISION DESIGN

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INTRODUCTION

The principle of elastic averaging has been known and practiced for eons [1, 2, 3, 4, 5, 6], but is often overlooked in the modern world of CAD where everything appears exact and more things are printed with the expectation of the same perceived perfection achieved by printing documents. A typical example is a gear designed with an excellent custom tooth geometry and printed for testing; however the bore is not quite accurate so it does not fit properly onto the shaft, so the young engineer tries to drill out the bore which shatters the plastic. Even in precision machined metal parts, all too often dowel pins are relied upon but the high stiffness of pins pressed into holes requires clearance, and as a result accuracy is less than it could be. On the other hand, the ubiquitous LEGOTM brick mates with exquisite precision even to bricks made decades ago. Indeed, these same principles have even been applied to MEMs devices [2]. Accordingly, this paper will explore some simple design concepts and first order engineering calculations to help designers utilize elastic averaging features in there designs, those printed and those machined.

DESIGN THEORY

Think of one of the classic epitomes of elastic averaging: the use of multiple blade flexures to support a high precision optical element. Indeed, any structure can be thought of a collection of springs of stiffness, k_i , and the connection between structures is no different. When two elements are brought together, the effective springs between them will be forced into equilibrium, hence some forces may be very high, causing local yielding. In addition, a geometric equilibrium will be achieved, which may or may not result in the desired position between elements. The key is to deterministically design the springs such that when forced together, no spring is over stressed, and the resulting relative position of the elements is within desired tolerance. Furthermore, the desired global stiffness of the assembly must be achieved.

The first thing to decide is the elastic averaging function there to align the parts which are then

clamped together so global stiffness comes from the clamped (e.g., bolted) joint? Or is the elastic averaging function there to align the parts and once aligned provide the global desired stiffness. The former would be for a structural assembly and the latter would be the case for a printed gear to be mounted on a shaft. This can be accomplished with the following thought process:

- Define the desired accuracy of the assembly (δ_A).
- Define the expected geometric mismatch of the parts (manufacturing error) (δ_m). This is also the overall amount that springs can be expected to have to deform to accommodate manufacturing errors.
- The factor N better performance required is the ratio $N = \delta_m / \delta_A$
- It is reasonable to assume that the errors will be randomly distributed and thus conservatively the number of springs required to "average out" the manufacturing errors will be $N_s = N^2$.
- Define the desired resulting stiffness of the assembled joint, k_a .
- Set the number.

DESIGN EXAMPLE

Fig. 1a illustrates a box with a cavity that has a mounting plate to seal the top portion. The mounting plate has a protrude feature that matches the cavity of the box with a slight clearance gap forms between the walls of the feature and the cavity. In order to maintain the gap within the desired tolerances, the mounting plate needs to be precisely located in the middle of the box. Instead of using a traditional arrangement of two or three dowel pins mating with slightly oversized holes (or a hole and a slot) to locate the mounting plate, the concept of elastic averaging can be used to precisely locate and constrain the top mounting plate with respect to the box.

Locating the mounting plate requires three Degrees-Of-Freedom (DOF) in-plane constraints, i.e., along the X- and Y-axis, and about the Z-axis. For three dowel pins in slightly oversize holes, or pushing against the edges of three slots, it is an

almost exact constraint design since one dowel pin constrains the motion along the X-axis while the remaining two constrain the motion along the Y-axis, and about the Z-axis, except when parasitic forces move the dowel pins away from the slot edges. However, having four dowel pins layout on the box as illustrated in Fig. 1a will be an over-constrained design, which can be remedied with elastic contacts to average out the errors. Refer to Fig. 1b, each elastic contact is constructed by a pair of flexure beams and the slot in between is used to slot the mounting plate onto the top of the box with each dowel pin as a guide.

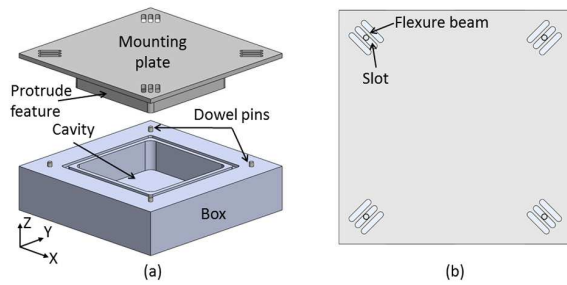


FIG. 1. (a) Illustration of a box with a cavity that has a mounting plate to seal the top portion. (b) Top view of the mounting top that has four symmetrical elastic contacts to locate the seal the box with four guiding dowel pins.

To constrain the mounting plate at a desired position, each elastic contact must provide precise constraint forces, which can be generated by deflecting the flexure beams with a fixed dowel pin. The approach is to press fit a dowel pin into a slot created by a pair of flexure beam. In this case study, the mounting plate is made of aluminum 6061 material with Young's Modulus of 71GPa and has an overall thickness of 3mm (excluding the protrude feature). The length and thickness of each flexure beam are 12mm, and 1.5mm respectively while the width is governed by the thickness of the plate. The focus of this case study is to understand the effectiveness of elastic averaging in averaging out the misalignment errors caused by the offset of the dowel pins due to manufacturing errors. The remaining sections present the theoretical model and the results obtained from both the analytical, and numerical analyses in predicting such misalignment errors.

MODELING APPROACH

The approach to predict the misalignment or deviation from the center of location is to first obtain

the stiffness model of the mounting plate with respect to (w.r.t.) its center. Then determine the total force vector and moment applied to the center of the mounting plate that could be due to the location offsets of the individual dowel pins. Using the Hooke's Law, the deviation can be obtained from

$$\begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} = [\mathbf{K}_{sys}] \begin{bmatrix} \Delta_x \\ \Delta_y \\ \theta_z \end{bmatrix} \quad (1)$$

Stiffness modeling of the top mounting plate

The mounting plate can be represented by a group of elastic contacts as illustrated in Fig. 2a. Each elastic contact is represented by a pair of springs that are tangential and normal to a dowel pin, which is fixed to the ground. The stiffness matrix of the tangential spring is represented as k_i^T and the normal spring as k_i^N where i denotes the numbering of each elastic contact.

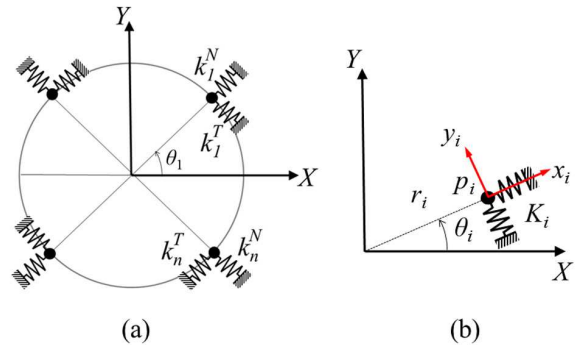


FIG. 2. Schematic of (a) a mounting plate with four equally spaced elastic contacts and (b) the orientation of the each flexure w.r.t. the global frame.

The global frame of the mounting plate is attached in the middle and each elastic contact has a local frame that is attached on the center of the dowel pin as shown in Fig. 2b. Here, p_i represents the local of each dowel pin while θ_i and r_i represent the orientation, and the distance of each local coordinate frame w.r.t. the global frame. For planar motion, the rotation matrix that describes the local frame w.r.t. the global frame is given as

$$\mathbf{G}_i = \begin{bmatrix} \cos\theta_i & \cos(\theta_i + 90^\circ) & 0 \\ \sin\theta_i & \sin(\theta_i + 90^\circ) & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (2)$$

Refer to Fig. 2b, the stiffness of each elastic contact, k_i^E , will be the sum of the stiffness matrices of the tangential and normal springs, i.e., $k_i^T +$

$\mathbf{k}_i^N$. Using Eq. (2), the compliance matrix of each elastic contact w.r.t. the local frame is obtained from

$$\mathbf{C}_i = \mathbf{G}_i(\mathbf{k}_i^E)^{-1}\mathbf{G}_i^T \quad (3)$$

From Eq. (3), the overall stiffness of each elastic contact w.r.t. the global frame is $\mathbf{K}_i = \mathbf{C}_i^{-1}$. Subsequently, the stiffness matrix of the mounting plate, $\mathbf{K}_{sys}$, is expressed as

$$\mathbf{K}_{sys} = \sum_{j=1}^n \mathbf{J}_i^{-T} \mathbf{K}_i \mathbf{J}_i^{-1} \quad (4)$$

where $\mathbf{J}_i$ is the Jacobian matrix given as

$$\mathbf{J}_i = \begin{bmatrix} 1 & 0 & -r_{y,i} \\ 0 & 1 & r_{x,i} \\ 0 & 0 & 1 \end{bmatrix} \quad (5)$$

where $\hat{\mathbf{r}}_i$ represents the position vector of each local frame w.r.t. the global frame.

Refer to Fig. 1, the four elastic contacts can be represented by the tangential springs, $\mathbf{k}_i^T$. Classical beam element can be used to determine the stiffness matrix of each tangential spring as shown in Fig. 3 and the stiffness matrix of a beam element in three DOF in-plane motion is given as

$$\begin{bmatrix} fu_1 \\ fv_1 \\ mq_1 \\ fu_2 \\ fv_2 \\ mq_2 \end{bmatrix} = \begin{bmatrix} \frac{EA}{L} & 0 & 0 & -\frac{EA}{L} & 0 & 0 \\ 0 & \frac{12EI}{L^3} & \frac{6EI}{L^2} & 0 & -\frac{12EI}{L^3} & \frac{6EI}{L^2} \\ 0 & \frac{6EI}{L^2} & \frac{4EI}{L} & 0 & -\frac{6EI}{L^2} & \frac{2EI}{L} \\ -\frac{EA}{L} & 0 & 0 & \frac{EA}{L} & 0 & 0 \\ 0 & -\frac{12EI}{L^3} & -\frac{6EI}{L^2} & 0 & \frac{12EI}{L^3} & -\frac{6EI}{L^2} \\ 0 & \frac{6EI}{L^2} & \frac{2EI}{L} & 0 & -\frac{6EI}{L^2} & \frac{4EI}{L} \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ q_1 \\ u_2 \\ v_2 \\ q_2 \end{bmatrix} \quad (6)$$

where E is the Young's Modulus of the material, I is the moment of area, A represents the cross-sectional area and L represents the length of the beam element. The sign conventions of the force, f , and moment, m , and respective displacement of each axis are shown in Fig. 3.

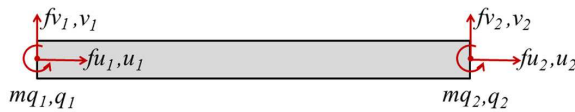


FIG. 3. Illustration of a beam element.

In an ideal condition, the dowel pin is constrained in the middle of each pair of flexure beams. Hence, it can be represented using four separate beam elements where the dowel pin becomes a constraint for each beam as illustrated in Fig. 4. Each beam has a frame attached to the end and

the a local frame is attached to the center of the dowel pin.

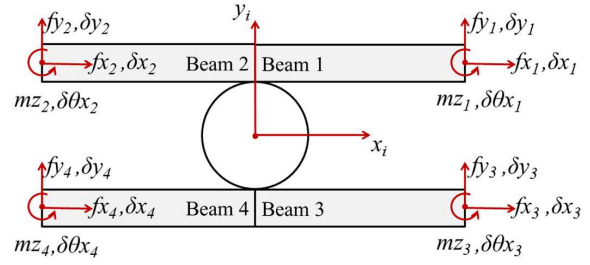


FIG. 4. Representing each elastic contact with four separate beam elements and their respective frames w.r.t. the center of the dowel pin.

By treating the the dowel pin as a support and each beam as a simply support beam, the maximum deflection of each beam can be determined by $\delta_{max} = \frac{fl^3}{48EI}$ where l represents the length of one entire flexure beam. By considering $L \leq l/2$, the stiffness matrix of Beam 1, $\mathbf{k}_{b1}$, and Beam 2, $\mathbf{k}_{b2}$, are expressed as

$$\mathbf{k}_{b1} = \begin{bmatrix} \frac{EA}{L} & 0 & 0 \\ 0 & \frac{6EI}{L^3} & -\frac{3EI}{2L^2} \\ 0 & -\frac{3EI}{2L^2} & \frac{2EI}{L} \end{bmatrix} \quad (7)$$

$$\mathbf{k}_{b2} = \begin{bmatrix} \frac{EA}{L} & 0 & 0 \\ 0 & \frac{6EI}{L^3} & \frac{3EI}{2L^2} \\ 0 & \frac{3EI}{2L^2} & \frac{2EI}{L} \end{bmatrix} \quad (8)$$

The stiffness matrix of Beam 3, $\mathbf{k}_{b3}$, will be similar to Beam 1, i.e., $\mathbf{k}_{b3} = \mathbf{k}_{b1}$ and the stiffness matrix of Beam 4, $\mathbf{k}_{b4}$, is same as Beam 2, i.e., $\mathbf{k}_{b4} = \mathbf{k}_{b2}$. Subsequently, the overall stiffness of each tangential spring, $\mathbf{k}_i^T$, is determined by summing up the stiffness of all four beams using

$$\mathbf{k}_i^T = \sum_{j=1}^4 \mathbf{J}_{bj}^{-T} \mathbf{k}_{bj} \mathbf{J}_{bj}^{-1} \quad (9)$$

where j denotes Beam 1 to 4 and the expression of Jacobian matrix, $\mathbf{J}_{bj}$, is similar to Eq. (5) except $\hat{\mathbf{r}}$ will be representing the position vector of the beam frame w.r.t. the local frame in this case. With the stiffness of each tangential spring determined, Eq. (2) and (3) are used to obtain the stiffness matrix of each elastic contact w.r.t. the global frame, $\mathbf{k}_i$. Lastly, the global stiffness of the mounting plate is obtained using Eq. (4).

Applied forces and moment

After obtaining the stiffness model of the mounting plate, the next step is to determine the total force vector and moment applied to the center of the mounting plate. There are two sets of forces that need to be determined for any elastic averaging problem. The first set of forces is a collection of forces generated by the deflections within the elastic contacts due to manufacturing errors such as the offset of the dowel pins etc. The second set of forces is a collection of equilibrium forces generated by the equilibrium condition of a multi-springs system.

As mentioned earlier, each slot between a pair of dowel pin is press fitted to a dowel pin to generate constraining forces as illustrated Fig. 5a. Hence, the flexure beams are represented as springs with the dowel pin being the ground and the forces generated by a pair of flexure beams are expressed as

$$\begin{aligned} F_a &= k_{22}[(g_d + \varepsilon) - g_o] \\ F_b &= k_{22}[(g_d - \varepsilon) - g_o] \end{aligned} \quad (10)$$

where g_o represents the original slot gap, g_d represents the slot gap after deflection, which is determined by the radius of the dowel pin, and ε represents the potential offset of the dowel pin due to the manufacturing errors. For F_a , k_{22} is the second diagonal component obtained from $\mathbf{k}_{b1} + \mathbf{k}_{b2}$. For F_b , k_{22} is the second diagonal component obtained from $\mathbf{k}_{b3} + \mathbf{k}_{b4}$.

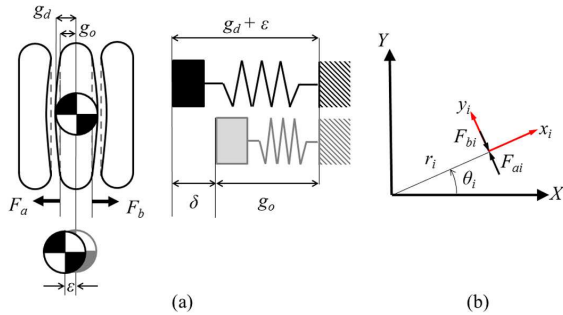


FIG. 5. Illustration of (a) different parameters between a beam and dowel pin that will generate the reaction force within an elastic contact.

Eq. (10) suggests that having a slot gap smaller than the diameter of dowel pin will generate a pair of constrained forces to constrain the elastic contact in a desired position. Most importantly, it accounts for any offset from the dowel pin, which will generate additional forces (from each pair of

flexure beams) that deviates from the desired position. Subsequently, the first set of force and moment vector w.r.t. the global frame is obtained from

$$\begin{bmatrix} F_{x1} \\ F_{y1} \\ M_{z1} \end{bmatrix} = \begin{bmatrix} \sum_{i=1}^n (F_{bi} - F_{ai}) \sin \theta_i \\ \sum_{i=1}^n (F_{ai} - F_{bi}) \cos \theta_i \\ \sum_{i=1}^n (F_{ai} - F_{bi}) \cdot r_i \end{bmatrix} \quad (11)$$

The next step is to determine the second set of force vector, i.e., the equilibrium forces. The equilibrium condition of each individual multi-springs system is unique. Using a simplified dual-springs system as an example, both springs are constrained by a rigid body with the dowel pins being the fixed ground. Like any form of multi-springs system, an offset, ε , from one of the dowel pins creates a new equilibrium position for the dual-springs system as illustrated in Fig. 6.

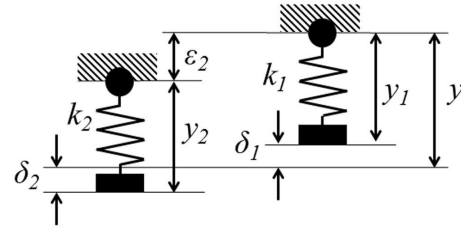


FIG. 6. Illustration of the simplified double springs system with one of the dowel pins having a position offset.

This new equilibrium position, y , due to the constraint of the rigid body and the equilibrium of forces is expressed as

$$y = \frac{k_1 y_1 + k_2 y_2}{k_1 + k_2} \quad (12)$$

Assuming $k = k_1 = k_2$ yields $y = (y_1 + y_2)/2$. To derive a model that is generic for four elastic contacts and more, it is much convenient to first assume that all elastic contacts will return to an equilibrium position similar to the given dowel pin offsets, ε . Refer to Eq. (12) as an example, such equilibrium position will be the sum of all position offsets divided by the total number of elastic contacts. Consequently, the balance displacement vector, $\mathbf{BD}$, which consists of the displacement of each elastic contact needs to recover so as to return to its equilibrium position, is given as

$$\mathbf{BD} = \mathbf{O}^T - \mathbf{E} \cdot \mathbf{O}^T \quad (13)$$

where $\mathbf{O}$ represents the vector of position offsets from the dowel pins and $\mathbf{E}$ represents the matrix

of equilibrium positions ($n \times n$), which is formed by a unit matrix with each component divided by the total number of elastic contacts.

Next, the balance displacement of each elastic contact obtained from Eq. (13) is used to determine the equilibrium forces. The expressions from Eq. (10) are used to determine these forces except that ε represents the balance displacement instead. Similarly, the expressions within Eq. (11) are used to obtain the second set of force and moment vector w.r.t. the global frame, $[F_{x2} \ F_{y2} \ M_{z2}]^T$. Subsequently, the final force and moment vector w.r.t. the global frame is

$$\begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} = \begin{bmatrix} F_{x1} \\ F_{y1} \\ M_{z1} \end{bmatrix} + \begin{bmatrix} F_{x2} \\ F_{y2} \\ M_{z2} \end{bmatrix} \quad (14)$$

Lastly, the deviation from the center of the mounting plate due to the position offset of the dowel pins are obtained using Eq. (1) and (14), and the global stiffness matrix obtained.

ANALYTICAL AND NUMERICAL RESULTS

The analytical model was used to predict the deviation of the mounting plate along the X- and Y-axis, and about the Z-axis, due to the offsets of the dowel pins' locations. Fig. 7 shows the mounting plate with numbering of respective elastic contact and offset sign convention of the dowel pin. It also provide the values of the offsets given to these elastic contacts that were used in the analytical analyses. Total of three runs were conducted based on three set of conditions.

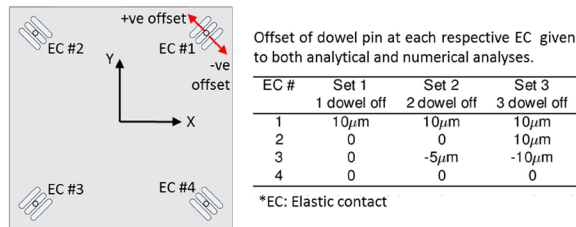


FIG. 7. Numbering of elastic contacts and the offset of dowel pin given to each contact.

In addition, Finite-Element (FE) analyses were also conducted via the ANSYS Workbench 16.0 using those similar sets of conditions. For each run, both analytical and FE analyses returned the deviations from the center along the X- and Y-axis based on the given dowel pin offsets. (Note that the angular deviation about the Z-axis was not

compared because ANSYS does not provide a direct function to obtain angular motion. Translational displacements can be easily obtained from the functions provided by ANSYS.)

TABLE 1. Analytical and FE results of the deviations from the center due to dowel pins offsets for a mounting plate with 4 elastic contacts.

	1 dowel pin off	2 dowel pins off	3 dowel pins off
Analytical			
Δ_x (m)	-2.63E-07	-3.94E-07	-7.88E-07
Δ_y (m)	2.63E-07	3.94E-07	2.63E-07
Numerical (averaged values across the center)			
Δ_x (m)	-2.50E-07	-4.34E-07	-8.72E-07
Δ_y (m)	3.04E-07	4.51E-07	3.15E-07
Variations			
Δ_x (%)	5.19	9.17	9.68
Δ_y (%)	13.72	12.61	16.64

The results obtained from the analytical analysis were compared against the results obtained the FE analysis and are listed in Table 1. Results show that the average variation between the analytical and FE results is about 11% with a maximum deviation of 16.6% is observed in Set 3. The analytical model is better in predicting the deviation along the X-axis but overall provides accurate prediction of the deviations based on first order modeling approach.

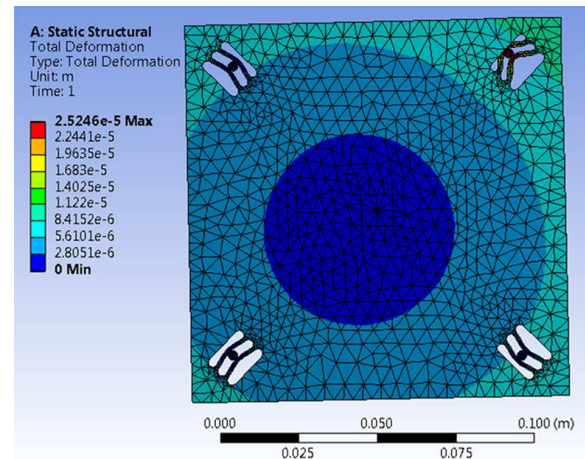


FIG. 8. FE model of the mounting plate due to the offset of one dowel pin in EC #1.

Fig. 8 shows the FE model of the mounting plate due to the offset of one dowel pin in EC #1. It provides a good illustration on how elastic averaging works. With the dowel pin in EC #1 having a positive offset of 10 μ m from the initial position,

it was observed that the remaining three elastic contacts also deflected. This because all elastic contacts deflected to the equilibrium positions and thus "average out" the offset of the dowel pin in EC #1. Hence, the deviations along the X- and Y-axis were not more than $0.3\mu\text{m}$. Refer to Set 3; three dowel pins off, in Table 1 again, the deviations were within $1\mu\text{m}$. These results show how elastic averaging works effectively.

The vacuum chuck assembly shown in Fig. 9 was created to enable a technician (later in production a robot) to pick up a flat rectangular battery separator coated with electrolyte paste located in a nest defined by three pins and transfer it to a stack whose element locations are also defined by three pins. The uncoated region of the separator is only 2mm wide, and the region coated with electrolyte cannot have a vacuum drawn on it or else the electrolyte liquid will pull through. Hence there is only a very narrow perimeter region for a vacuum to act to pick up the element. To make an economical efficient vacuum gripper, which may later need to be made in large quantities for a factory, a minimum number of parts and machining steps are required, where the primary functional requirement is that the sides of the gripper must contact the three pins that defines the stack position before the element is lowered onto the stack.

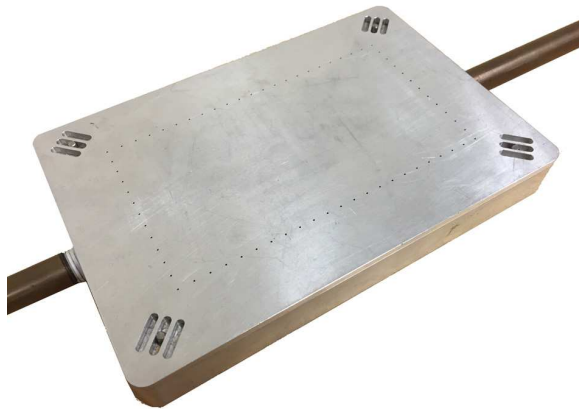


FIG. 9. Vacuum chuck assembly.

The most efficient design was found to be one where the base with vacuum distribution channels is machined along with four holes for dowel pins. The 3mm thick vacuum surface top plate that contacts the part to be acquired is made on a precision waterjet machining center (OMAX MicroMAX <https://www.omax.com/omax->

machine/micromax) which cuts the many orifices required as well as the flexures to mate with the dowel pins and the precision edges that are located with respect to the orifices and the flexures. A double sided adhesive tape is then applied to the base, and cut out were there are vacuum distribution channels. The top plate is positioned on the tops of the dowel pins projecting from the base and pressed into place. The result is that on the precision located side surfaces, there is no detectable edge discontinuity. Previously without using the elastic averaging locating method described above, when attempting to lower a top plate onto a base held against the three locating pins, some human jiggling always resulted in misalignment.

CONCLUSION

Elastic averaging has worked well throughout history to help create precision machines. The use of defined "rigid" pins and "elastic" flexural elements was the key then, and the key now. With the advent of rapid manufacturing methods from abrasive waterjet machining, laser cutting, and 3D printing, surfaces, complex features, such as many flexures designed to mate with many dowel pins, are easily created for very low cost. Hence assemblies of parts made by these processes can be as precise as those made from conventional milling, turning, and grinding processes.

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NOVEL DESIGN AND FABRICATION APPROACH FOR LIGHT WEIGHT HIGH NATURAL FREQUENCY MACHINE TOOL STRUCTURES USING PRECISION REPLICATION

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INTRODUCTION

A novel manufacturing and design technique for building carriages, gantry beams and slide tables for machine tools has been developed and field proven for many years in PCB Mechanical via drilling and routing systems. This new manufacturing and design technique called **Vacuum Replication Fabrication (VRF)** has the following advantages over traditional weldment and casting manufacturing techniques:

- 1) Nearly zero internal stresses developed during the fabrication process
- 2) Cost effective, a typical structure will cost 30% to 50% less than a traditional weldment or casting
- 3) Very short lead times for fabrication, usually 50% to 72% faster than traditional weldment or casting fabrication processes
- 4) Typically no need for additional grinding of precision surfaces required after lamination
- 5) Structures that are designed and fabricated with the VRF technique tend have high resonant frequencies compared castings or weldments with the same form factor.

Description of the fabrication technique

The VRF technique utilizes a composite construction approach, where elements are glued together using structural epoxy. To achieve precision surfaces, a novel replication technique that involves vacuum pressure is used to push the surface skins of the structure flat against a granite surface plate (Figure 1, Figure 2 and Figure 3). Once the skin is held down to the granite, the internal structural elements (e.g. thin wall tubing) are attached to the skin using structural epoxy. Once the epoxy cures, the vacuum pressure is released. The parts lift off the granite with a net flatness that is almost identical to the granite surface. The above process is repeated to fabricate the other

structural subassemblies, then all of the subassemblies are laminated together to make the entire structure.

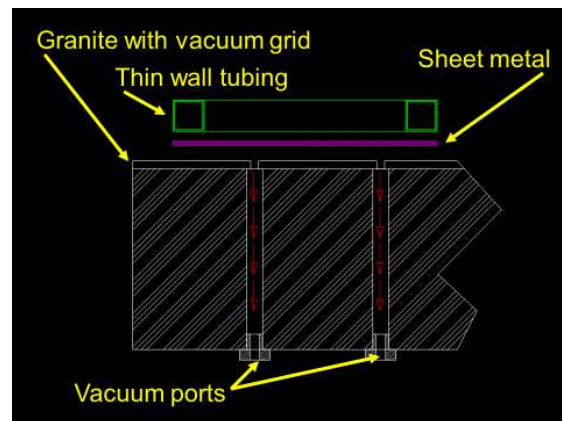


FIGURE 1. Cross section schematic showing how the vacuum is channeled through the granite to push the sheet metal against the granite

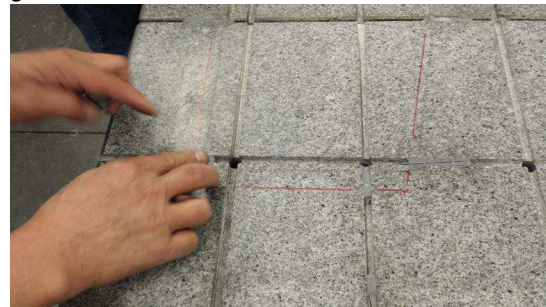


FIGURE 2. Close up of the vacuum replication granite showing the grooves and holes that the vacuum pulls through



FIGURE 3. Example of a structure's skin vacuumed against the granite

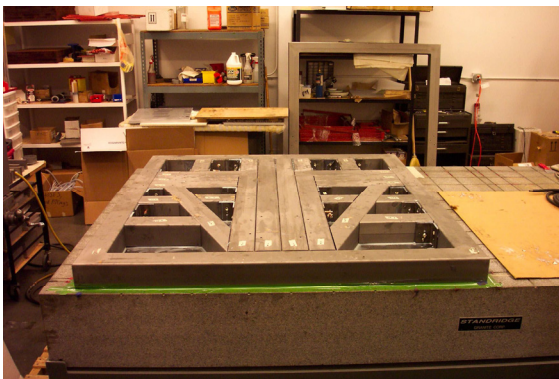


FIGURE 4. Example of a structure subassembly after the epoxy has cured and the subassembly is about to be lifted off the Granite

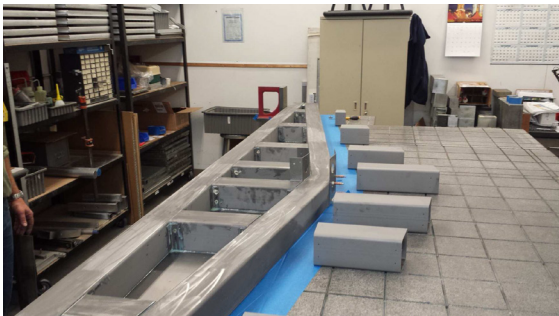


FIGURE 5. Example of a structure subassembly after the epoxy has cured and the subassembly is about to be lifted off the Granite

Examples of slide tables and Gantries made using this process:

Figure 6. A PCB mechanical drilling system Y axis slide table measuring 2800mm x 762mm x 66mm (110"x 28.5"x2.5") with a flatness and parallel deviation of 0.075mm (0.003") between the two 2800mm x 762mm surfaces. Total mass of the slide table 130 Kg (286 lbs.)

FIGURE 7. PCB mechanical drilling system X/ Y axis gantry measuring 4000mmx500mmx200mm (157.5"x19.68"x8.0")

with a mass of 242Kg (532 lbs.) with geometric surface tolerance of 0.012 mm.



FIGURE 6.



FIGURE 7.

Example of cost savings and lead time improvements:

Table 1 below compares the fabrication cost and fabrication time of the slide table in figure 6 that is made using traditional welding and grinding practices versus the VRF process.

Fabrication method	COST USD	Weeks to fabricate
Welding & grinding	\$5500 USD	8 weeks
VRF Technique	\$3300 USD	3 weeks

Table 1. Cost and lead time comparison of the VRF process vs. a weldment process

The main reasons why the welded assembly cost more and takes longer to fabricate is due to the need for stress relief and grinding after welding. The actual welding time is also longer compared to gluing parts together.

The design aspect of the VRF process

The VRF design process is a fusion of machine tool design and aircraft design. Aircraft design tends to lean towards light structures and stiffness is usually a secondary concern, where machine design puts stiffness first and foremost and weight as a secondary requirement.

This VRF design fusion came to be when Linear motors replaced ball screws in PCB drilling systems. Linear motor drives are significantly more sensitive to payload than ball screw drives, every Kilogram counts in the design of the linear motor drive moving mass while stiffness is still a paramount requirement. The beauty of the VRF method is that the manufacturing process does not put any restrictions on the wall thickness of the structure and one can vary the structures material thickness where it is best needed or not needed to obtain the best balance of stiffness and weight. As in Composite design, there are infinite possibilities with the core material when using the VRF method. The PCB drilling system application utilized thin wall steel tubing as a favorite core material. Below is a short list of core materials used in VRF

- Thin wall tubing
- Honeycomb
- Additive manufactured structures
- Machined out webs from plate material
- Metal foam

Skin material is usually cold rolled steel or aluminum. Typical sheet thickness that works the best is 1.5mm to 2mm thick. The thickest skin material that has been used in the VRM process is 4mm. Once the material gets thicker than 4mm other replication methods are required, this is due to the ability of the material to resist the vacuum pressure pushing it against the granite. An example of replicating thicker materials is shown in Figure 8, where an 11mm thick piece of precision ground stock is bolted against a precision ceramic face.



FIGURE 8. Alternative replication technique for thick material applications, using a precision ceramic fixture to attach the thick steel bars that will replicate the ceramic surface.

How does the epoxy bond line effect structures:

In the early days of vetting out the VRF process, understanding how the epoxy bond line affects the structure was an item that needed to be understood. The concerns about the bond line are listed below:

- 1) Epoxy Bond line effect on structures stiffness
- 2) Epoxy shrinkage and its effect on replication accuracy
- 3) Which epoxy types are best for the VRF technique
- 4) long term dimensional stability of the epoxy bond line
- 5) Long term robustness of the epoxy bond line
- 6) Bond line creep

The answers to these concerns are below, all of this information comes from experimental results

- 1) Epoxy bond stiffness has been tested by comparing two beam sample sets. The sample sets are made from the same stock length of rectangular steel tubing. One sample set is made up of two sections of thin wall tubes that are laminated together with epoxy. The other sample set is made up of two beam sections that are "stitch welded" together. The results of this test are shown in Figure 9. The data in Figure 9 shows that the laminated beam has better stiffness than the stitch welded sample set. The stiffness improvement of epoxy over

the welded sample was 14% to 16% better. This stiffness improvement is attributed to the large cross sectional area that the epoxy covered vs the “stitch welded” cross sectional area.

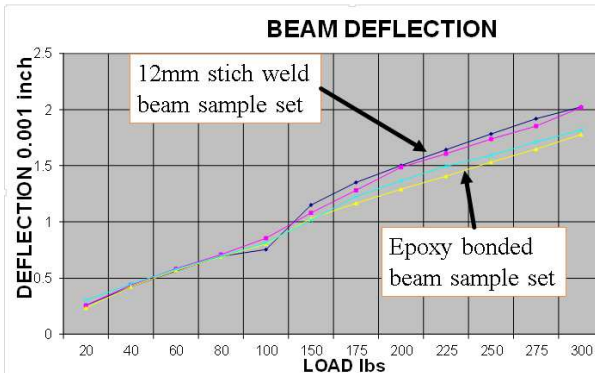


FIGURE 9. Load Vs. Deflection curves comparing welded beam and bonded beams.

- 2) Typical epoxy shrinkage is 1% of the bond line thickness. Keeping bond line thickness between 0.4 mm to 0.2mm is very important when tight surface tolerance is need to be held. But with more typical VRF applications, bond lines up to 0.6mm can be tolerated.
- 3) The best epoxy type for the VRF technique is Structural epoxies, especially ones that are “rubber toughened”. These rubber toughened epoxy are very robust and resist impact delamination very well.
- 4) VRF assemblies that have been deployed in the field for many years have been evaluated for dimensional accuracy and have been found to hold there tolerance in the harsh environment of a production floor
- 5) With over 350 system in the field, some of them are 15 years old. Not one report of bond line delamination.
- 6) As a rule of thumb, if the total stress in the bond line is kept below 3.44 MPa (500psi), bond line creep is not an issue in certain structural epoxies.

An example of performance of a VRF structure

The light and stiff aspects of the structural elements in a VRF design lend themselves to a final structural assembly that has high resonant frequencies.

An example of this outlined below:

This VRF assembly is an XY gantry that supports an X axis assembly that holds 6 individual Z axis assemblies. The dimensions of the VRF gantry assembly is 4000mmx500mmx200mm (157.5”x19.68”x8.0”) with a mass of 242Kg (532 lbs.). The total mass of the gantry assembly with the X axis and 6 Z axis assemblies is 369Kg (812 lbs.) and has a first mode of 154 Hz. This first mode is illustrated in Figure 10. The completed gantry assembly is shown in figure 11 mounted to the system base.

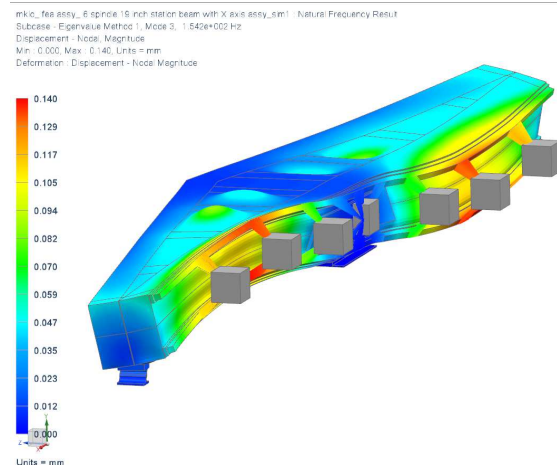


FIGURE 10. FEA Model of first mode of the gantry at 154 Hz. Spindle assemblies and x axis carriage modeled as lumped masses attached to the gantry.



FIGURE 11. XY Gantry Shown assembled on the system

FEA Modeling for VRF structures:

A considerable amount of Modal testing and FEA modeling have been done over the years. The primary reason for this FEA to actual Modal comparison is to find the best FEA representation of the epoxy bond line. It turns out that making FEA bond line elements with the same mechanical properties as the elements that are getting bonded together, yields FEA results that are very close to the modal results of the actual structure, as seen in Table 2

Struture Mode	DP460 epoxy	stitch welded	FEA	Percent deviation
1	110 Hz	108 Hz	131 Hz	16%
2	342 Hz	344 Hz	354Hz	3.40%
3	648 Hz	646 Hz	676Hz	4.10%
4	718 Hz	706 Hz	680 Hz	5.00%

Table2. Test results comparing Modal data of welded and laminated beam samples to FEA analysis results

Conclusion

The VRF manufacturing and design technique is field proven method that enables machine tool designers to improve the performance of a system and decrease to cost of the system at the same time.

A NEW IMPROVED COUNTERFORCE MECHANISM FOR VERTICAL MACHINE TOOL AXES AS APPLIED TO A LARGE OPTICS GRINDING MACHINE

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Background

Recent developments in machine tool technology have achieved higher accuracy and dynamic performance by, among other things, replacing ballscrews with linear motors. The improvements are largely due to the elimination of compliant mechanisms placed between the controlling component (the motor) and the component to be controlled (the carriage).

It is generally not possible to apply linear motor technology to vertical axes unless some means of counteracting the gravitational force of the moving components is implemented. Linear motors have no mechanical advantage, therefore the force required to maintain axis position against gravity is high, as it is directly applied.

For a linear motor to generate a high force, high electrical currents are necessary, with substantial heat generation the result. Heat is generally recognized as being one of the largest contributors to errors in machine tools.^[1]

Furthermore, the cost of large linear motors is high, as is the power consumed continuously.

Couterforce Mechanisms Various

Historically, there have been numerous methods applied to counterbalance the mass of vertical axes, and Cranfield Precision have, over the years, tried many.^{[2],[3]} Simple counterbalance designs typically comprise a vertical carriage moving on the front face of a structure, with a counterbalancing carriage (of equal mass) mounted to the rear or sides of the structure. The carriage and the counterbalance are connected via chains, cables or belts, over pulleys at the top of the structure.

This method is acceptable where low velocities and accelerations are required, but serious problems emerge when the enhanced dynamic performance demanded by modern systems is attempted.

The moving mass and therefore the force and power required for acceleration is doubled, and, when tight dynamic control is attempted it becomes clear that the system effectively

comprises two masses, connected by an undamped spring - with little control over the second mass. This severely limits the ability to maintain good servo control of the axis.

A later design replaced the counterbalance carriage with a pneumatic cylinder, with a pressure regulator set to balance the cylinder force with the carriage mass. The cylinders are necessarily of large diameter and often display stick-slip characteristics. In addition, the flow of air to cater for rapid carriage movements is high, so large valves and reservoirs are required. Hysteresis in the valve leads to variations in force on reversal. This reduces machine accuracy.

An illustration of the reduction of machine accuracy due to the shortcomings of a vertical axis is illustrated in Figure 1

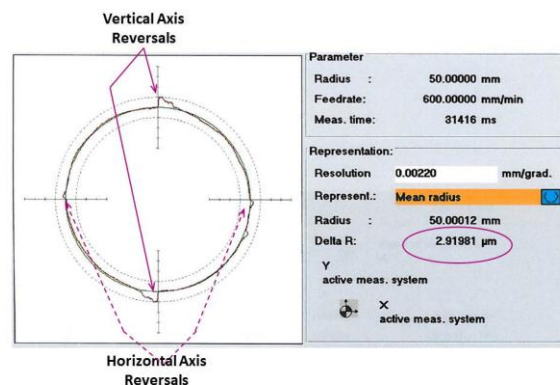


Fig. 1. Circular Interpolation Errors at Reversal on a Machine with a Vertical and Horizontal axis.

As can be seen, deviations from the perfect circular interpolation are observed at the 4 reversal points. In this example, the magnitude of the errors are three times higher for the vertical axis ($3\mu\text{m}$) than the horizontal axis ($1\mu\text{m}$). This measurement was taken on a machine when new. Wear in the servo valves controlling the counterbalance air cylinders has since increased the errors, and along with an

increase in the stick-slip in the cylinders, led to the need to make costly replacements. Hydraulic equivalent designs are also available, but suffer from similar problems, in particular even greater friction in the piston seals.

The New Method.

The basic elements of the new method are shown in the diagram, Figure 2. The machine (primary) carriage is guided by linear bearings in a conventional manner, such that it is free to move vertically. The primary drive, a linear motor, is positioned in the normal method, although not necessarily capable of resisting the carriage mass.

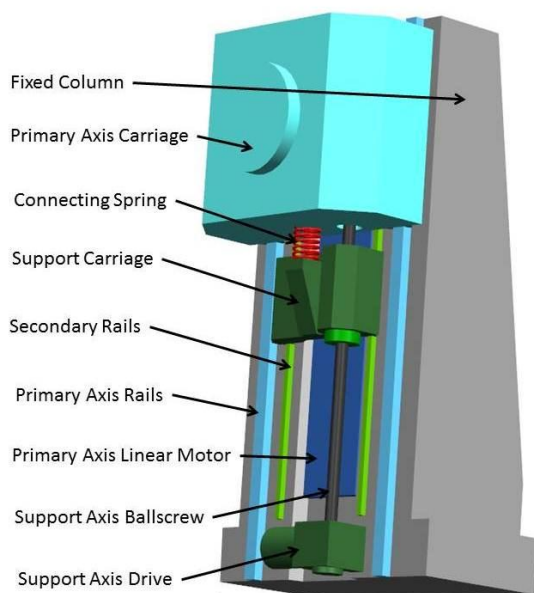


Fig. 2. New Counterforce Concept

A second axis is now introduced, parallel to the primary axis, being a conventional screw driven device having its own guideways, and an independent position measuring system. The function of this secondary axis is to act as support for the primary axis, the supporting force being transmitted through a spring.

In the concept design shown above, the support axis is placed below the primary carriage and acting at the centre of gravity. Other options are to use tension springs and hang the carriage from them, or to place the support carriage within the primary carriage – as is the case with the machine described below.

During the initial machine power-up, the secondary axis is initiated first. It then rises so the spring contacts the primary carriage, which is then lifted after some spring compression. At

this point, the mass of the primary axis is fully supported by the spring, and primary drive system is initiated.

The primary drive can now hold carriage position with little to no force required. From now on, all moves commanded for the primary axis are made with a simultaneous, equal move by the secondary axis, and therefore the distance between the primary and secondary axis is maintained, and thus the spring compression remains constant. Note, there is no cross communication between the two axes when the system is running normally.^[4]

Test Rig

A simple rig was put together to test the principle (Fig 3). This comprised an air bearing slide with linear motor and scale, and a ballscrew driven secondary slide, all items being available at the time. A bracket was attached to the secondary slide with a spring positioned between the two carriages. Despite the test rig shortcomings the principle was shown to work extremely well.



Fig. 3. Test Rig

Features of the New Method

A vertical axis configuration, as described above, has been built into a precision Optics Grinder, and has been shown to display dynamic position errors of the same magnitude as a horizontal axis of equivalent mass. This has not been possible previously in the experience of Cranfield Precision.

This is because the primary linear motor is now able to apply the forces necessary to overcome the inertia of the primary axis and maintain a high level of dynamic positioning, without the need to supply an additional, large continuous force merely to overcome gravity.

Another major attribute of the system is that the only force being applied to the primary carriage is the counterforce, via the spring, which is positioned to apply this force at the centre of mass of the carriage. Any misalignment between the primary and secondary axes, that could potentially disturb the axis motion of the primary carriage is negated, as the only connection between the primary and secondary carriage is through the spring, which displays very low levels of lateral stiffness.

The spring's rate also needs mentioning. This should be high enough to apply the necessary force, while being compliant enough to minimize the effects of positioning errors from the screw driven axis. For the example described, a vertical carriage of 800 kg is supported by a spring that compresses 40mm when supporting the load. Should the secondary axis have positioning errors of 0.1mm, (which is likely for a low cost screw-driven axis), the variation in counterforce will be $0.1/40$, or 0.25% of the 800kg, ie. 20N. This is not a significant force for a motor sized to control position of an axis of this mass where inertial forces must still be overcome.

Should an upward bias be required however, for example to move a tool away from a component during a power failure, this is achieved simply by adding an offset value to the relationship between the primary and secondary axes by a fixed amount.

Re-establishing a high level of balance is also simple if, for example a heavy tool is attached to the vertical axis, as the system can be re-balanced in moments to cater for the change in mass.

There are two methods that can be applied to do this. The simplest method is to momentarily disable the primary axis, allow the equilibrium to return, and re-establish the offset. This may not be practical in some cases, where control of the primary axis position cannot be lost at any time. If this is the case, the counterforce can be corrected by observing the primary drive torque (current) and reducing it to zero by adjusting the position of the secondary carriage such that the spring compression is changed. Either method can be automated via the CNC, instigated by a tool change for example, or a manually requested operation.

There could also be cases where a known, constant force, possibly process related, is applied to a vertical axis, where an adjustment of the counterforce would be beneficial.

The temptation to make this a dynamic in-process background task has been avoided, however, as the potential to de-stabilise the system in certain conditions is high.

There are other advantages with this system, in particular when considering safety, and the application of brakes. On power down, particularly un-planned, there is no dramatic failure. A simple screw-mounted clamp, or motor-mounted brake will hold both the primary and secondary axes in position, with little force required, due to the screw's mechanical advantage.

Applying the Design to a Precision Optics Grinding Machine.

The method has been applied, among other applications, to the vertical axis of optics grinders recently built by Cranfield Precision^[5].

The machines, for precision grinding optical components currently up to 1.6 metres diameter, comprise a horizontal X axis, travelling rotary table, and a vertical axis, fixed bridge mounted, which carries the grinding wheelspindle. The total mass of the moving vertical axis is in excess of 800 kg. Figure 4.

The error budget generated during the concept design for the machine highlighted the vertical axis as being by far the most critical for precise dynamic position control, as non-axisymmetric components are required to be machined in, among others, an R- Θ mode.



Fig. 4. Precision Optics Grinding Machine.

An overall view of the bridge and vertical axis assembly is shown in Figure 5.

Figure 6 shows the vertical axis assembly without the bridge casting displayed. Here the primary carriage, bearings and rails are visible, along with the two linear motors fitted either side

of the carriage. The two motors are driven in torque/follower mode. The measuring scale (not visible in the diagram) is mounted behind the bearing rails.

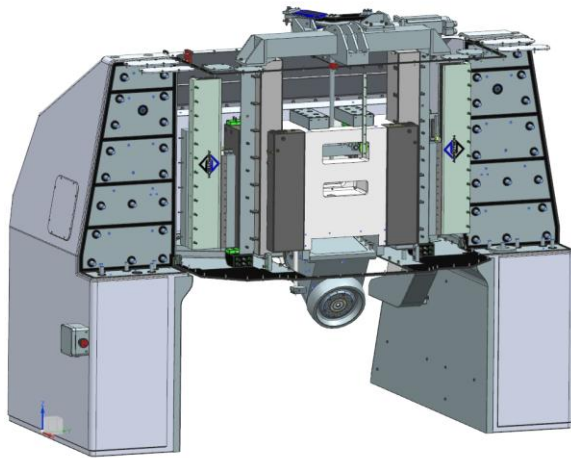


Fig. 5. The Machine's Bridge Assembly.

Figure 7 shows the counterforce mechanism alone, showing the springs, which are mounted on a separate secondary carriage positioned within the primary vertical carriage.

It should be noted, that this secondary carriage is mounted to its own, independent guide-way, whose task it is to fully constrain the secondary carriage in all but the vertical direction.

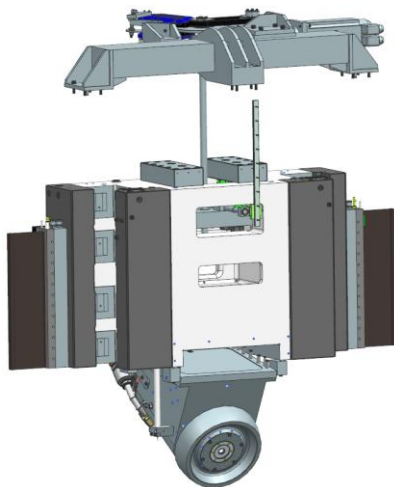


Fig. 6. The Machine's Vertical Axis Assembly.

As a result, any disturbing forces, from, for example reaction forces from the torque applied to the ballscrew are not transferred to the primary axis.

Also shown is the ballscrew and drive, the latter in this case incorporating a right-angle reduction

drive to increase the mechanical advantage of the counterforce drive.

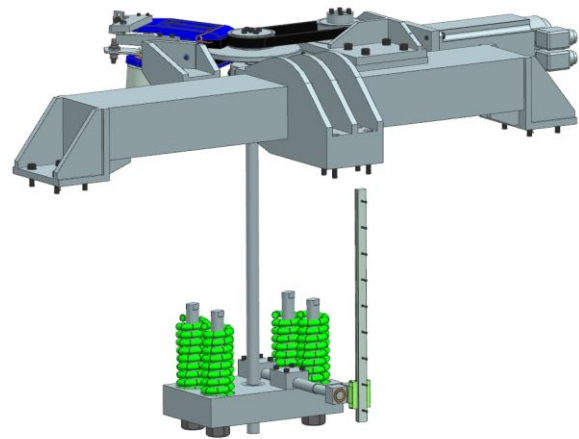


Fig. 7. The Counterforce Mechanism.

As a result, the secondary drive motor size is reduced, while remaining capable of achieving the speed required to match the primary axis velocities.

For the grinding machine shown, a small upward bias for the counterforce is preferred. As a result, in the brief moments between the servo drive disengaging and the clamp being applied, the axis moves upwards slightly, and away from the extremely high value workpiece.

As previously mentioned, the positioning accuracy requirement for the secondary drive is low, due to the forgiving nature of the low spring rate selected, so the position feedback provided by an on-motor encoder, and the introduction of a gearbox is not a restriction.

A rotary brake unit is fitted directly to the top of the ballscrew.

Axis Drive Performance.

To demonstrate the vertical axis' dynamic performance, a Ø250 mm part has been ground, of "saddle" form, comprising 15mm vertical saggital undulations at the outer edge. This was cut at 25 rev.min^{-1} (2 x undulations per rev, $25 \text{ rev.min}^{-1} = 0.84\text{Hz}$). The vertical carriage mass was 800kg, and the velocity bandwidth exceeds 60Hz. During the move, observed following errors do not exceed $\pm 0,5 \mu\text{m}$. Figure 8 displays the position errors during the cut, showing the reduction in following error as the cut moves towards the part centre - where the height of the undulations, and errors, approach zero.

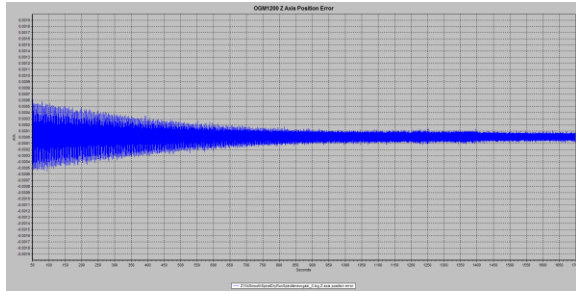


Fig. 8. The Vertical Axis Dynamic Performance

Cost Benefits

Initial perceptions are that application of this technique has added a complete servo axis to the machine, therefore increasing the cost. Assuming however that the advantages of applying linear motor technology is required, and the option of fitting a large linear motor that can hold the axis against gravity is rejected, a counterforce mechanism of some sort must be considered.

All the parts for the mechanism described have been identified, and their full cost established, and compared to the mechanism used a few years ago when pneumatic cylinders were applied to apply counterforce to a carriage of approximately similar mass.

Running costs are also reduced, due to the lower power requirements for the linear motor, and inefficiencies in the cooling system.

Summary

Applying this counterforce method to a precision machine tool vertical axis can realize a vertical axis that can achieve the dynamic positioning capability of a horizontal axis, with the addition of a low accuracy and low cost ballscrew-driven axis using rolling element guideways, and a proprietary motor/gearbox/encoder package. Cost to implement the method is low, as compared to other, less accurate methods. Thermal loads on the machine are also kept to a minimum.

Acknowledgements

Thanks go to John Fitzpatrick and the Design, Assembly and Commissioning teams at Cranfield Precision for turning the concept into a successful machine.

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LASER AUGMENTED DIAMOND DRILLING OF HARD AND BRITTLE MATERIALS

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ABSTRACT

Cost of a final product in the precision engineering is relatively high due to the level of accuracy and caution that is needed. In high precision machining, this high cost is mainly attributed to the high level of accuracy of the cutting tool and equipment, machine down time for tool set and tool replacement, low material removal rate (MRR) and precision and etc. Developing new processes that overcome these obstacles and at the same time improve the quality of the final products is always a demand. Hard, brittle and difficult to machine materials such as ceramics, semiconductors and composites have a variety of applications in precision engineering industry and often they need to be drilled to a high level of accuracy. Due to their mechanical properties, low fracture toughness and high strength, it is very challenging to drill holes free of fractures, surface and subsurface damages, cracks and micro-cracks and with good edges and high surface quality. Their high hardness causes rapid tool wear that not only decreases the drilling efficiency, but also increases the down time of the process. Therefore, the cost of drilling these types of materials is usually high and processes available for this purpose are expensive.

There are many drilling techniques available to generate precise micro/macro holes in hard and brittle materials such as conventional and mechanical methods [1-2] and non-conventional processes. Nonconventional methods are such as electrical discharge machining, waterjet, micro-electrical chemical machining and ultrasonic machining [3-8]. However, some of these processes are limited in cost, machining efficiency, dependence on the material properties, and achievable aspect ratio. Laser drilling (ablation) is another process for making holes in these materials as it is a non-contact process and able to make a hole in almost any material. However this process suffers from high power consumption, size inaccuracy, tapered

holes, redeposited material, micro cracks due to thermal stresses and heat affected zone [9-10]. The Laser Augmented Diamond Drilling (LADD) is an under development technology for drilling of hard and brittle materials. In this process a laser beam is focused through an almost transparent tool/tip (usually diamond) to drill the workpiece. The LADD process utilizes high compressive and shear stresses induced by a diamond tool/tip and laser heating to thermally soften, thus reducing the hardness and brittleness of the workpiece material. Significant cost reduction is possible as the LADD can increase the MRR, decrease the tool wear, improve the edges and surface finish, and minimize or eliminate sub-surface damages occurring during the drilling operation. An ideal schematic of the LADD technique is shown in Figure 1.

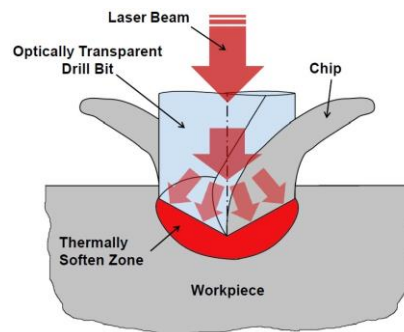


Figure 1: Schematic of the LADD process

EXPERIMENTAL SETUP

The laser used in this study is an IR CW fiber laser with the wavelength of 1070nm. In this process laser should be focused on the cutting edge of the tool and should just increase the temperature of the material in touch with tip of the tool, therefore a medium power laser unit is sufficient.

The cutting tool used for drilling operation, is a single edge diamond drilling bit with a 0.5 mm radius shown in Figure 2. Laser beam is in Gaussian profile and it is tried to cover as much

as possible of the cutting edge. A reconfigured Universal Micro-Tribometer (UMT) was coupled to the LADD system to perform the drilling tests. For this work, to be able to use the laser, sample is rotating instead of the tool, which is similar to drilling operation in turning process. For this purpose, samples were mounted on a precise air bearing spindle.

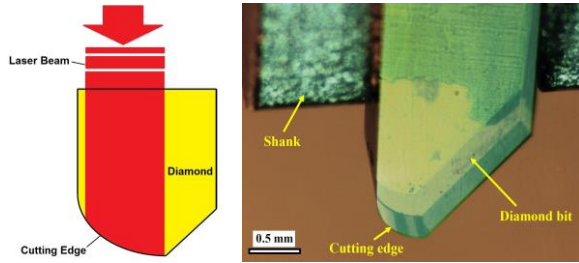


Figure 2. Diamond bit used in LADD

Achieving high accuracy and small tolerances and avoiding or minimizing the residual stresses due to cutting or clamping, which can cause post process distortion to workpiece or damage during process, are some essentials in precision manufacturing. Therefore, it is recommended that workpiece should be cut with lowest possible forces. Lowering cutting forces is also beneficial for the tool life which increases the repeatability of the process and decreases the deviation of the tolerances of the final products. Therefore measuring the cutting forces can help to clarify the cutting mechanism during the process and it can lead to improve the quality of the drilled holes. Since cutting forces have the major effects on the tool wear, monitoring them gives a better understanding of the process. By comparing the forces obtained in different laser power conditions, the effect of thermal softening on the cutting and therefore tool life can be evaluated. For this purpose a Kistler dynamometer which is capable of measuring forces up to 250N is utilized in the experiments. The dynamometer is based on piezoelectric technology and is stiff enough in the setup. The LADD system was mounted on dynamometer to be able to measure the forces during the tests as shown in Figure 3. Output signals were collected by National Instrument Data acquisition (DAQ) card and processed by the Matlab software.

EXPERIMENTAL PROCEDURE

In this study, tests with different conditions of the laser power, feed rate and type of cutting fluid were carried out. Table 1 shows the parameters

and their levels. The lower level of the feed rate, $2\mu\text{m}/\text{rev}$ is based on maximum reported chip thickness in literature [11] for the silicon (100), 120 nm, to achieve a ductile mode cut. Laser powers are also the output powers measured with a power meter before each tests. Even though the focus of this study was on the benefit of using LADD to decrease the thrust forces, the quality of the holes especially the entrance was under consideration too. Due to the size and shape of the drill bit, which has a negative rake angle, dimple shape blind holes were generated. Two type of cutting fluids were used in these tests, DI water as a water based and Odorless Mineral Spirits (OMS) as an oil based cutting fluids. As the laser beam is going through the diamond bit and to the cutting edge, it will not be interrupted by cutting fluid. The material used in this study to perform the tests, was single crystal silicon (100) as the research group has many experiences in working with it.

Table 1. Parameters and their levels

Parameter	Value
Laser Power	0, 4, 12 W
Feed Rate	2, 4 $\mu\text{m}/\text{s}$
RPM	1000 RPM
Max. depth of the holes	300 μm

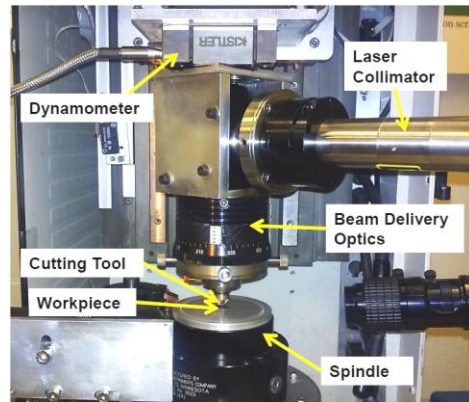


FIGURE 3. LADD setup

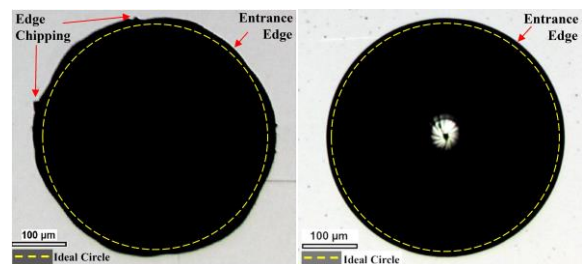


FIGURE 4. Edge entrance quality before and after process optimization

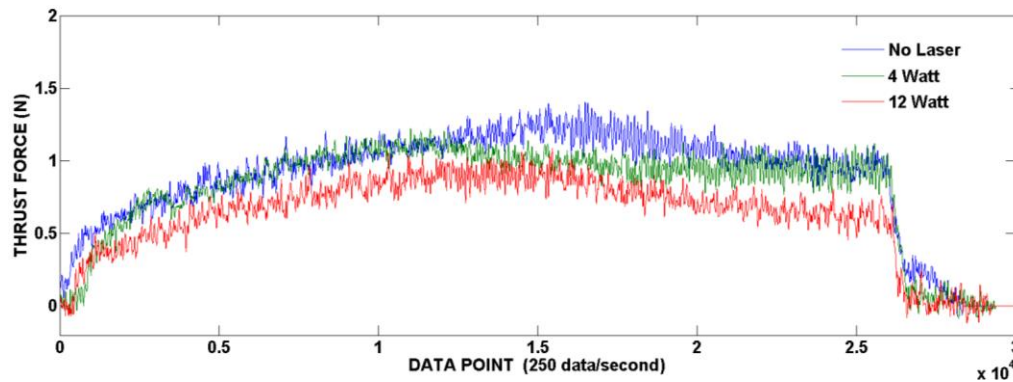


FIGURE 5. Thrust forces obtained in the test with $2 \mu\text{m/s}$ feed rate and DI water as cutting fluid

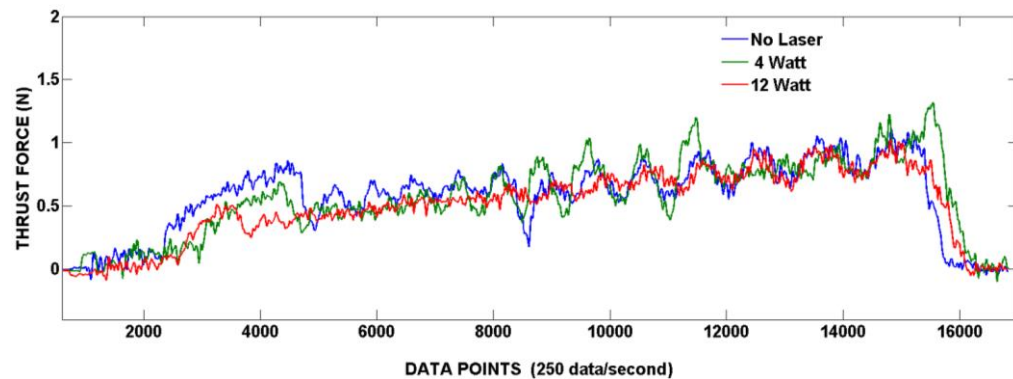


FIGURE 6. Thrust forces obtained in the test with $4 \mu\text{m/s}$ feed rate and DI water as cutting fluid

Drilled samples were imaged and the quality of the entrance edges was analyzed. Initial results show that by using proper laser power and adjusting other parameters, in optimum condition desirable results can be achieved. Figure 4 shows the entrance edge quality before and after using proper parameter levels.

CUTTING FORCES

Thrust forces were collected during the experiments, while the motion of the drill bit toward the workpiece was in control based mode. Figure 5 shows the thrust forces acquired during the test with $2 \mu\text{m/s}$ feed rate and DI water as the cutting fluid. Using 4W laser power slightly decreased the forces; however a higher laser power, 12W, lowered the forces $\sim 30\%$. The graph of the thrust forces for the test with $4 \mu\text{m/s}$ feed rate and DI water shows lower forces compare to previous test condition ($2 \mu\text{m/s}$ feed rate). It is due to change in the cutting mode from mainly ductile to brittle. It is well known that in brittle mode condition forces are lower as fractures happen during the cut and make the material removal easier. Figures 7 and 8 show the bottom of the holes drilled with these two feed rates. In the case drilled with $4 \mu\text{m/s}$ brittle mode cut, black areas, is dominant. The high

deviation of the forces in graphs in Figure 6 is suggesting that fracture happened in the entire process from beginning. The entrance of the hole is not showing any sign of sever fracture at the edge.

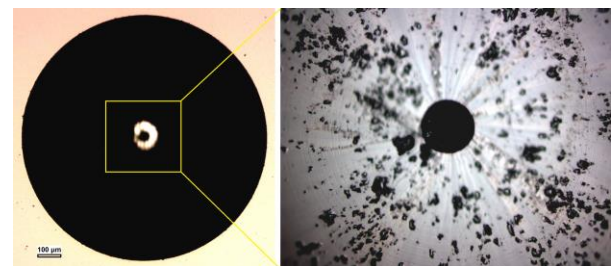


FIGURE 7. Microscopic image of bottom of the hole drilled with $2 \mu\text{m/s}$ feed rate

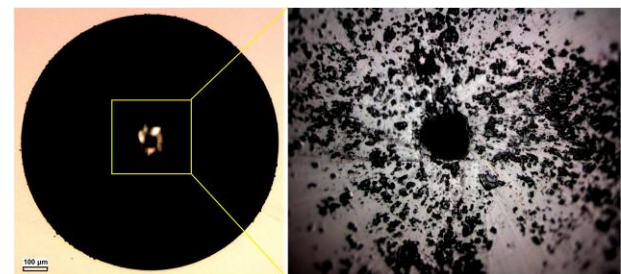


FIGURE 8. Microscopic image of bottom of the hole drilled with $4 \mu\text{m/s}$ feed rate

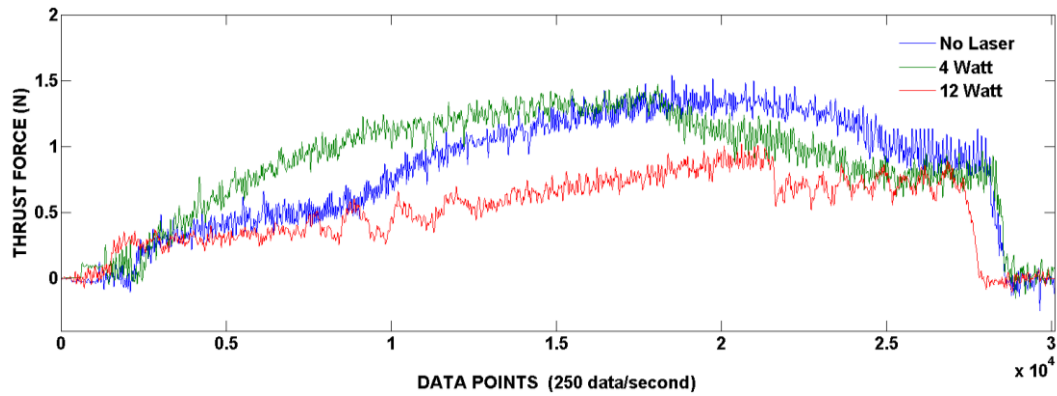


FIGURE 9. Thrust forces obtained in the test with $2\ \mu\text{m/s}$ feed rate and OMS as cutting fluid

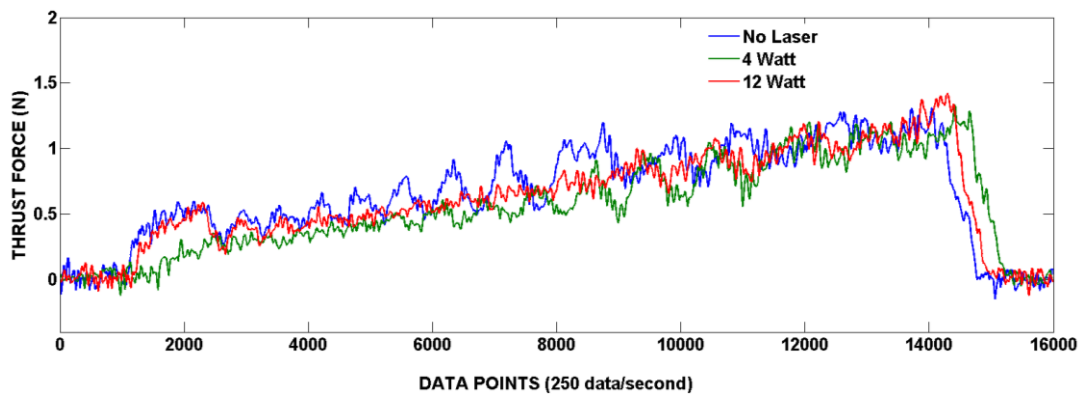


FIGURE 10. Thrust forces obtained in the test with $4\ \mu\text{m/s}$ feed rate and OMS as cutting fluid

Thrust force graphs for drilling tests with aid of OMS, as the cutting fluid, is also show the same trend as with the DI water. Increasing laser power for the tests with $2\ \mu\text{m/s}$ feed rates decreased the forces as shown in Figure 9. Figure 10 shows as cutting is dominated in brittle mode, no obvious difference can be seen for different laser power conditions. Although changing laser power is not showing any significant effect on thrust forces in $4\ \mu\text{m/s}$ feed rate for both cutting fluids, a close look at the edges shows that by using laser a better edge quality with less edge chipping compare to no laser case can be achieved, Figure 11.

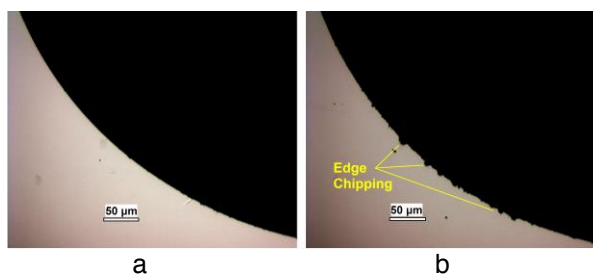


FIGURE 11. Edge quality at $4\ \mu\text{m/s}$ feed rate
a) 12 W b) No laser

In machining of the brittle materials, achieving smooth surfaces with minimum fracture and cracks on the surface is in high priority in precision engineering applications. On the other hand, lower cutting forces leads to less tool wear and therefore fewer tool replacements. Fewer tool replacements not only decreases the tool cost, but also decreases the down time of the equipment.

EFFECT OF CUTTING FLUIDS

Evaluation of the thrust forces when different cutting fluids were tested, suggests that using DI water decreases the forces slightly compare to OMS, Figure 12 and 13. It is probably due to embrittlement effect that water can cause and make the material more brittle and easier to cut. Comparing the graphs in Figure 12 and 13 also shows that using laser made the graphs smoother (less oscillation), at least at first half. Less oscillation in forces means a more stable force condition, which will benefit both tool and workpiece.

CONCLUSION

LADD is a new technique to drill hard and brittle materials by combining mechanical drilling and

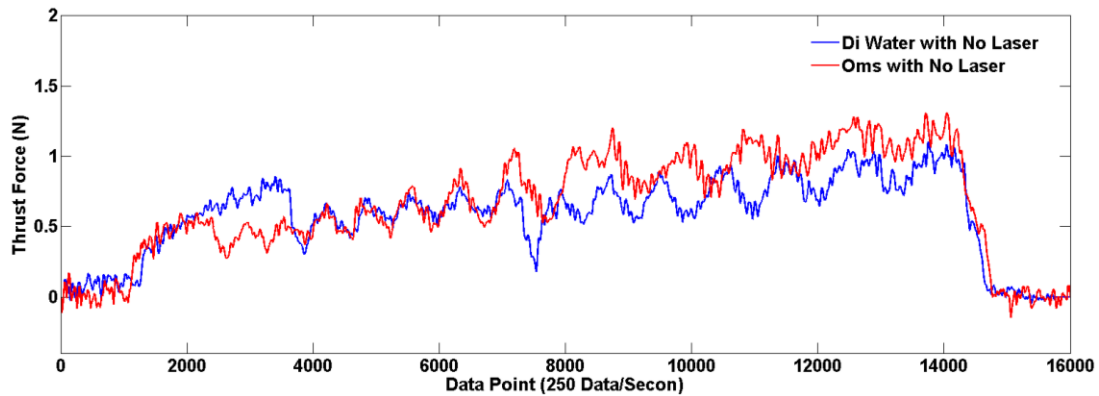


FIGURE 12. Thrust forces obtained in the test with $4 \mu\text{m/s}$ feed rate, No laser

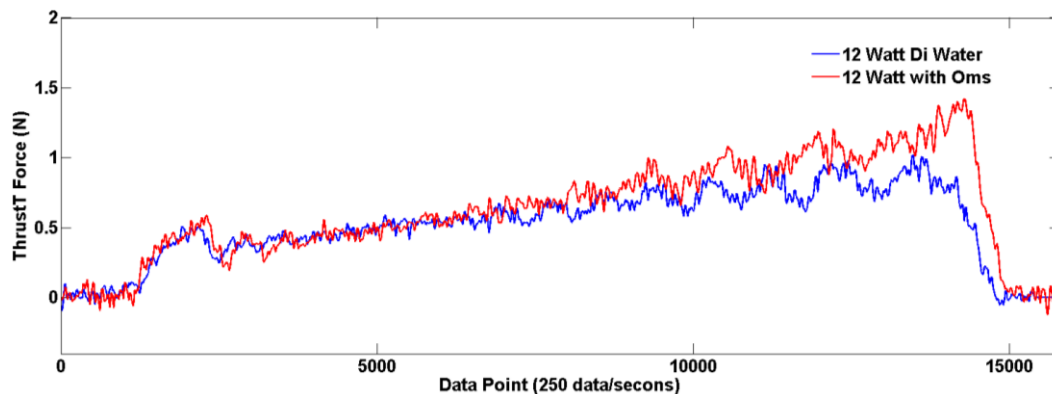


FIGURE 13. Thrust forces obtained in the test with $4 \mu\text{m/s}$ feed rate, 12 W

laser heating. This hybrid process decreases the required thrust force and therefore save the cutting edge of the drill bit for a longer time and tool will last longer. This tool life increase not only decreases the cost of the final product but also improves the quality of products and repeatability of the process which means more identical products.

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Numerical and Experimental Investigation of Two-Phase Forming of Polymeric Biconvex Lenses by Droplet Printing

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INTRODUCTION

Optical lenses are widely used and studied due to their broad applications. Mainly, there are three types of manufacturing methods, i.e. direct subtractive machining[1], molding[2], and additive manufacturing[3][4][5]. Both direct machining and molding require sophisticated facilities. Additive manufacturing (AM) method is recently developed. The most studied AM method for microlenses is drop on demand (DOD)[3]. DOD method deposits one or several liquid droplets on a substrate and forms lenses. DOD method shows numerous merits including low cost, simple process, good surface finish, *etc.* However, this method also has the limitation of one-side curvature. In previous study, we demonstrated a method for manufacturing polymeric biconvex lenses using multiphase printing at a water-air interface[4]. Surfactant was added to adjust interfacial tensions. The influences of drop volume and surfactant concentration to lens shape were experimentally studied. Biconvex lenses with good surface finish can be obtained.

In this method, interfacial tension is one of the most important factors of lens forming. There are many studies on evaluating interfacial tensions and contact angles by using the already known droplet shape at multiphase [6][7][8][9]. However, none of them is on the calculation of lens shape from forming condition and most of these studies ignored the influence from gravity and buoyancy. For the multiphase printed microlens, it is important to predict the shape of the lens before fabrication. Obtain the designed lens shape by controlling the printing parameters, such as material properties, volume, interfacial tensions, *etc.*, is also critical to this process. For large size lenses, gravity and buoyancy also played an important role and cannot be ignored. In this work, we aim to develop a numerical method to calculate the lens shape considering both micro scale and

macro scale forces, valid the numerical method through experimental tests and study the influence of different forming parameters.

NUMERICAL SOLUTION

When a monomer droplet is deposited on an immiscible liquid and forms a lens, force balance of this monomer droplet on the liquid surface is shown in Figure1a.

At equilibrium, the gravity of the monomer lens, $\vec{G}$, needs to balance the buoyancy, $\vec{F}_B$, and interfacial tension force, $\vec{F}_S$. The force balance equation is shown in Eq. 1. In this study, we use Young-Laplace (Y-L) equation and momentum Navier-Stokes (N-S) equation as governing equations. Eqs. 2-5 are derived from these two equations in coordinate o_1xz (Figure1b) for the upper curve. Similarly, Eqs. 6-9 are derived in coordinate o_2ry (Figure1c) for the bottom curve.

Eq. 10 is Neumann's quadrangle in horizontal direction[10]. Interfacial tensions can be measured in advance, while line tension can be calculated using an empirical equation. We cited Neumann's study on line tension calculation and add an empirical coefficient k , which is presented in Eq. 13 [11]. k can be decided through the test. Eq. 11 is the macro-scale force equilibrium equation in vertical direction. Eq. 12 is the mass conservation equation of the lens. Based on these equations, along with the boundary conditions, lens profiles can be calculated. Two coordinates and the cross section of three phases at contact line are schematically shown in Figure1. The program flowchart is shown in Figure 2. Matlab was used for the numerical calculation in this study.

$$\vec{G} = \vec{F}_S + \vec{F}_B \quad (1); \quad \frac{dx}{ds} = \cos \varphi \quad (2);$$

$$\frac{dz}{ds} = \sin \varphi \quad (3); \quad \frac{d\varphi}{ds} = 2H_u + \frac{z}{\lambda_u^2} - \frac{\sin \varphi}{x} \quad (4);$$

$$\begin{aligned} \frac{dv}{ds} &= \pi x^2 \sin \varphi \quad (5); & \frac{dr}{dp} &= \cos \vartheta \quad (6); \\ \frac{dy}{dp} &= \sin \vartheta \quad (7); & \frac{d\sigma}{dp} &= 2H_b - \frac{y}{\lambda_b^2} - \frac{\sin \vartheta}{r} \quad (8); \\ \frac{du}{dp} &= \pi r^2 \sin \vartheta \quad (9); \\ Y_u \cos \varphi + Y_b \cos \vartheta - Y_w \cos \alpha + \sigma R &= 0 \quad (10); \\ 2\pi R Y_w \sin \alpha + \rho_{0b} u g - \rho V g &= 0 \quad (11); \\ V &= v + u \quad (12); \\ \sigma &= -k \cdot Y_u \frac{d \cos \varphi}{d \kappa} \quad (13); \end{aligned}$$

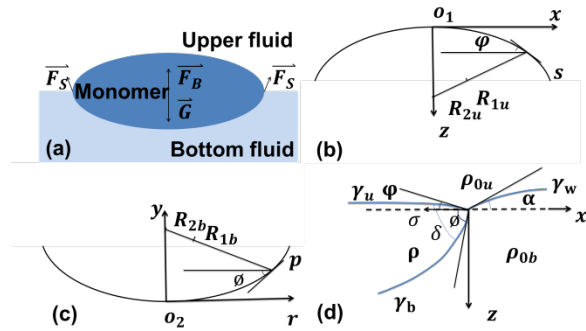


FIGURE 1. Schematics of force balance and coordinate systems. (a) Force balance of a monomer droplet on liquid surface, (b) upper curve, (c) bottom curve, (d) Cross point of three interfaces and cross-section.

In Eqs. 2-13, x, z, r, y are horizontal and vertical coordinates of the upper curve and the bottom curve, s, p are arc lengths of the upper curve and the bottom curve, φ, ϑ are the tangential angles of the upper curve and the bottom curve, and v, u are volumes of the upper portion and the bottom portion, respectively. V is the total droplet volume. R is the radius of the lens at static state, ideally, R equals to x and r at static state. Y_w, Y_u , and Y_b are the interfacial tensions of the water (water/air interface), the upper curve (monomer/air interface), and the bottom curve (monomer/water interface), subscripts w, u and b represent the water phase, the upper and the bottom portion, respectively. σ is line tension at the multiphase (air/monomer/water) line. κ is the curvature at the contact line. $\rho, \rho_{0u}, \rho_{0b}$ is the density of the monomer, the air and the water, respectively. We neglected the buoyancy of air in this study. H_u and H_b , are the mean curvatures at the origin of coordinates (o_1, o_2). λ_u and λ_b are the capillary numbers of the system for the upper and bottom portion,

respectively. Mean curvatures and capillary numbers can be calculated by [6]

$$H = \frac{1}{R_1} + \frac{1}{R_2} \quad (14); \quad \lambda = (\rho - \rho_0)g/\gamma \quad (15);$$

where R_1, R_2 are the principal radii of curvature, which are shown in the Figure 1 b-c, and g is gravity constant.

At first, we input material properties, including the interfacial tensions of monomer/air (γ_u), monomer/water (γ_b), and water/air (γ_w), the densities of monomer (ρ), air (ρ_{0u}), and water (ρ_{0b}). We also input the volume of the droplet, an estimated upper portion volume and angle, and then start the iteration process. The calculation is based on the Young-Laplace equation and Navier-Stokes equation, which has stated in Eqs. 2-9. All possible combinations are calculated and saved after the iteration. The combinations are selected based on the force balance and mass conservation using Eqs. 10-13. All the possibilities which meet all the equations are saved and the average and standard deviation is automatically calculated and exported as an output. Based on the mean parameters, the predicted lens shape can be calculated and plotted.

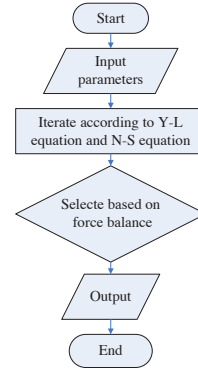


FIGURE 2. Program flowchart.

EXPERIMENTAL VALIDATION

To validate the numerical model, experiments were conducted. A syringe (10ml, BD) and flat end precision stainless steel needles (Nordson EFD) were used to produce monomer droplets. Droplets of different volumes were created to investigate the influence of volume to the lens shape. A mixture of two monomer materials, Triallyl-1,3,5-triazine-2,4,6-trione (TATATO) (Sigma-Aldrich) and Pentaerythritol Tetra(3-mercaptopropionate) (PETMP) (Sigma-Aldrich), was used as droplet materials. The basic properties of the materials are shown in Table 1.

Different amount of surfactant (detergent) was added to Deionized (DI) water to adjust the interfacial tensions of monomer/water interface and water/air interface. To avoid the influence of material shrinking during solidification, we use the shape of the liquid lens to validate the numerical model in this research. Future research on the effects of material shrinkage will be studied. In this research, two sets of validation tests were conducted. For each set of test, the concentration of surfactant was fixed and we changed the volume of droplets. The interfacial tensions were measured using a CSC DuNouy interfacial tensiometer. Measured interfacial tensions of these two sets of tests were listed in Table 2.

TABLE 1. Material properties

Composition	Density (g/cm ³)	Refractive Index	Ratio (V %)
TATATO	1.159	1.510	0.6
PETMP	1.3±0.1	1.531	0.4
Mixed	1.177±0.019	-	-

TABLE 2. Interfacial tension in the experiment

	Surfactant Concentration (ml/l)	Interface tension (dyn/cm)		
		γ_u	γ_w	γ_b
Test 1	0.01	42	56.6	20.6
Test 2	0.02	42	44	18.4

RESULTS AND DISCUSSION

Figure 3 shows the relationship between lens shape and volume for both experimental and numerical results. The numerical results agreed with the experimental results and the difference may come from errors such as interfacial tension measurement and water evaporation caused interfacial tension variation. From Figure 3a, the lens radius increases with the increasing of volume. The numerically calculated radius was slightly larger than the experimental radius. While Figure 3b shows the numerical results of angles are slightly smaller than the experimental results. These two figures mean the numerically calculated lens shape is a little flatter than the experimental lens shape.

In Figure 3a, we compared the radius from test 1 and test 2. It shows that under the same interface tension, the lens radius increases with the increasing of droplet volume. By comparing blue points to green points, or comparing red points to purple points in Figure 3a, we can have the conclusion that under the same droplet

volume, the lens radius decrease when the surfactant concentration increases (interfacial tensions decrease). Figure 3b suggests that with the increase of the droplet volume, angles won't change too much. However, with the increasing of surfactant concentration (from test 1 to test 2), angles increase rapidly. Figure 3 shows that the lens size is influenced by the droplet volume, while the lens shape is strongly affected by the surfactant concentration. This result agreed with the previous research[4]. From the validation tests, it can be seen that the numerical model can well predict the shape of lens which is formed under multiphase condition.

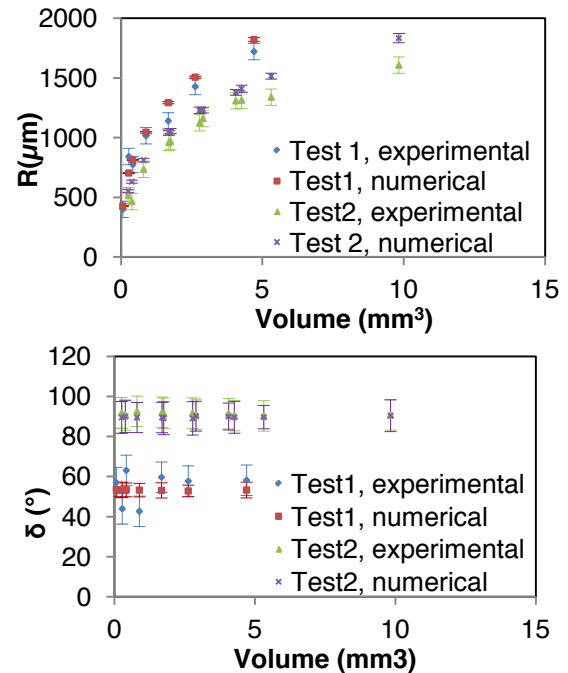


FIGURE 3. Validation and comparison. a). relationship between radius and volume, b). relationship between angle and volume,

Conclusion

The numerical model we presented in this work can calculate the shape and size of the lens which is formed using a multiphase printing method. In this numerical model, we considered both micro scale force balance by using Y-L equation and the N-S equation, and macro scale force balance of gravity, buoyancy, and interfacial tension. Experimental tests with different droplet volumes and surfactant concentrations have been conducted to validate the numerical model. The results validated that this numerical model is able to calculate the lens shape. From numerical model, it can be seen

that the lens shape is sensitive to the surface tension. This gives us a large tuning range for adjusting lens shape but on the other hand it needs precise control of interfacial tension to achieve designed lens shape. For future work, the influence on manufacturing process and the effects of lens shrinkage will be studied.

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TRENDS IN CONTROL TECHNIQUES FOR PRECISION MECHATRONIC SYSTEMS

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INTRODUCTION

The objective of this manuscript is to survey and report on trends in mechatronics and control systems of interest to the precision engineering community. These often involve motion control of some type, and in general we see efforts moving towards improving dynamic performance and the ability to model and predict complex dynamic behavior. The ASPE held a topical meeting on Precision Mechatronic System Design and Control in April 2016 [1] that included technical presentations and a tutorial [2] that highlighted these trends. A recent issue of *Mechatronics* focused exclusively on the control of high-precision motion systems [3], and manuscripts presented at the ASPE annual meeting also show evidence of these trends. We see three main trends standing out: (1) a growing recognition of the importance of feedforward control, (2) a continued use of primarily frequency-domain techniques with complementary data-based and model-based design techniques, and (3) an increasing use of distributed sensing and control - often using low-cost microcontrollers and open-source software.

FEEDFORWARD TECHNIQUES

Feedforward control techniques are increasingly important in achieving performance requirements in motion control systems. These techniques are often neglected, or presented as the open-loop techniques with substantial limitations which then necessitate feedback control. However, it is better to consider feedforward with equal importance to feedback control as a complete two degree of freedom design, with feedforward used to provide tracking of reference trajectories, and feedback to compensate for unknown disturbances, plant variations, and uncertainties. Feedforward control techniques either apply a prefilter on the reference trajectory, or use a model to precalculate the effort (the current in electromechanical systems) that will be required to move the stage through a given profile. A combined feedforward and feedback controller is shown in Fig. 1. Feedforward

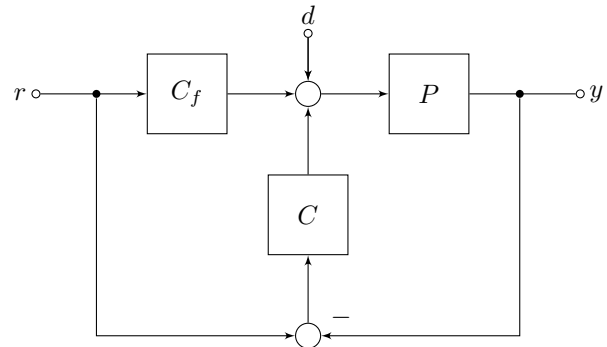


FIGURE 1. Block diagram of combined feedforward and feedback control. The reference (r) is directly filtered by feedforward gain C_f , and the error between output (y) and reference (r) is filtered by the feedback controller C . Disturbances (d) combine with the control effort before entering the plant P .

can be much faster than feedback control (feedback requires an error before it can act, feedforward does not), does not require any additional sensors or actuators, and does not create instability. It is also immune to sensor noise, but can amplify the effects of any discontinuities in the reference trajectory.

The most commonly used feedforward technique adds a control effort proportional to commanded velocity and acceleration, and generally act to invert the predicted response of an inertia acted on by viscous friction. These "Vff" and "Aff" terms are available on any number of commercially-available motion controllers. For systems comprising flexures an additional position dependent term compensating the static spring force is common. Boerlage et al [4][5] demonstrated tracking error improvements by using higher-order terms, namely jerk-derivative (or snap) feedforward, to compensate for the effect of higher-order dynamics at low frequency. Limiting the feedforward corrections to low frequencies allows the feedforward model order to be lower than is the order of the

controlled plant. More recent is work by Boeren, Oomen, and Steinbuch [6] in which they recast the conventional feedforward terms as an FIR filter with coefficients that are updated after every move based on the measured performance.

Feedforward control is discussed throughout this annual meeting of the ASPE. Tutorials by Ruijl and Ludwick, and by Okwidire and Sencer both discussed different techniques. Sato and colleagues present their work on feedforward augmented with a learning control element to improve tracking errors on an ultra-high acceleration linear motor stage [7], and Ryan presents modeling techniques that guide the feedforward filter selection [8]. Feedforward filtering is closely related to the concept of trajectory generation, and many so-called tuning problems are often best solved by modifying the position command. Often times feedforward techniques, which generally involve a number of derivative, or backward-difference calculations, are of limited effectiveness when presented with non-smooth trajectories. Tajima and Sencer present their work on interpolation techniques that create jerk-limited profiles thus enabling the effective use of feedforward [9]. For extremely fast but small corrections, large power density piezo actuators in combination with feedforward techniques were investigated by Hoogers [10]. The predictability of the used Lithium Niobate shear piezoelectric actuator (LiNbO_3) allows high accuracy feedforward piezo self-sensing techniques.

We are also seeing a convergence between different implementations of feedforward filters. Araki [11] presents an excellent tutorial-style overview of two degree-of-freedom control structures (feedforward for reference tracking, and feedback for disturbance rejection), and in particular, demonstrates the equivalence of the different forms. The two most familiar to most users he calls the *feedforward type*, where in the outputs of the feedforward and feedback control blocks sum with the disturbance to create the plant input, and the *set-point filter type* which takes the form of a filter exclusively on the reference command. In practice, the feedforward-type block is often used for an approximate model inversion, while the set-point filter is used for vibration suppression via notch filters or input shaping on the reference. Boeren [12] presents a unified process for tuning both, and demonstrates the results on a magnetic bearing stage. Iwasaki et al [13] highlight

two degree-of-freedom control in their survey paper on high-precision motion control techniques, and give a specific example of the techniques applied to a galvano (galvo) mirror scanner system. Structural flexibilities are sometimes neglected in feedforward design, but Dirksen and colleagues presented a technique for designing a feedforward controller for a precision motion control stage that explicitly accounts for position-dependent compliance in the machine structure [14]. This is an excellent example of a mechatronic approach that enables higher performance by using algorithmic sophistication where purely mechanical solutions (stiffer but heavier mechanical designs) are unacceptable.

FREQUENCY-DOMAIN DESIGN

Frequency-domain design techniques, a subset of classical control, continue to enable the design and analysis of feedback controllers. There is little to suggest that they have met their performance limits, and are being applied even to multivariable controls problems. Butler [15] demonstrates how a coordinate transformation effectively recasts a highly-coupled multiple-input, multiple-output six degree of freedom lithography stage as a series of single-input, single-output classical control problems. These he solves to meet performance objectives posed by the lithography process. State-space techniques complement the classical control tools. At this annual meeting, Dumanli and Sencer [16] demonstrate a tuning strategy for non-collocated systems wherein the measurement is taken at the load side - which is generally closer to the point of interest than is the conventional drive-side measurement. A state-space optimal control technique is used to set closed-loop pole locations and speed of response, while a complementary frequency-domain presentation demonstrates associated stability margins.

One main feature of frequency-domain techniques is that they do not require a traditional mathematical model of the system (e.g. a transfer function or state-space representation) in order to be used. Many commercially-available motion controllers now contain tools for measuring frequency responses, making experimental frequency response data readily available. As demonstrated by Turhan [17] It is even possible to estimate a system model using strictly in-process measurements, eliminating the need to interrupt the manufacturing process for a system identification operation. This has led to

recent interest in data-based controller design methodologies to exploit this capability [18]. Also new is some recent work by Hoogendijk [19] that demonstrates how a data-based root locus plot (containing transfer function poles and zeros) can be experimentally derived from frequency-response data with the justifiable assumptions of marginal stability and lightly-damped mechanical resonances. Frequency domain techniques are generally based in the assumption of system linearity. Mainly, that the response is largely independent of input amplitude. This is valid for large (meso-scale) moves in most motion control systems, but the dynamics of nanoscale motions are often governed by a much different set of relationships. We are now starting to see frequency responses used specifically to measure nonlinear behavior. Yoon and Trumper measured the frequency response of a servomotor at multiple current levels, used the measurements to fit the parameters of a Generalized Maxwell-Slip (GMS) friction model, and then used that model in a feedforward friction compensation scheme [20]. In addition, Nuij and Rijlaarsdam present a series of tutorial-style articles that demonstrate the use of frequency-domain techniques, and build to an analysis of the types of nonlinearities commonly seen in motion systems [21] [22] [23]. An overview of different modeling approaches for nonlinear systems is presented showing which effects they capture, partially capture or do not capture is shown in Tab. 1

We see a number of uses of frequency domain analyses in this conference. Wang and colleagues present their work on mapping time domain performance specifications (which is how motion system specification are usually provided) into frequency-domain requirements on the control bandwidth [24]. This ultimately can be used to specify constraints on the mechanical design of the system. Mashrafi et al [25] present promising results on using the frequency content of error signals to identify and correct for thermal drift in X-ray microscopy. This is again a frequency-domain approach wherein the controller is designed to be insensitive to very low-frequency drifts that are characteristic of thermal errors, but still remains responsive to higher-frequency content in the reference signals.

MICROCONTROLLERS IN PRECISION MECHATRONIC SYSTEMS

On the one hand we see controllers for high-end systems become more complex to achieve

increasing levels of system performance by compensating non-ideal mechanics and physics. On the other hand, consumer electronic products, catalyzed by the maker movement, make available lower cost, lower performance hardware and software that is being used for robots, drones, 3D printers and similar mechatronic systems, according to Peijnenburg et al [26]. For robotics its common practise to use microcontrollers as opposed to dSPACE and xPC-like solutions in the mechatronic field. We are at the point that Arduino-like microcontrollers can be used to control a mechatronic test set-up, that Inertial Measurement Units (IMU) developed for phones and drones can be used to log acceleration levels in equipment, and that sensors for computer vision are cheaper than ever before. The low cost and availability of microcontrollers also enables distributed control of complex mechatronic systems. For instance Lu and Chen [27] proposes a fully flexible motion approach, omitting the repeated installing and uninstalling going from one process to the next. A part is installed on a stage once and combined they will travel through a number of processing stations until becoming an end product. Such a new approach demands one technology that allows herds of independently controlled motion stages to share an extensible working area. To circumvent disturbing dynamics as a consequence of attached umbilical cord to supply electricity, multiple independently and flexibly controlled motion stages are presented working in a shared planar region.

Finally, we see new opportunities for education in controls and mechatronics enabled by the availability of low-cost hardware. We may have reached a point where discrete-time control techniques, or at least implementation techniques, should displace some of the classical continuous-time approaches as normally taught in a first undergraduate controls course. Trumper has designed low-cost systems for magnetic levitation [28] and atomic-force microscopy [29] that, coupled with a myRIO real-time target from National Instruments, are highly suitable for teaching precision mechatronics. Low-cost hardware is also enabling hands-on controls education through online methods. White and Steinmeyer have developed a pair on classes through MIT and on the EdX platform that use an Arduino-controlled propeller to teach introductory controls [30][31]. All of the hardware components are readily-available to order online. Python is used to communicate

Legend: effect captured / info available effect not captured / info not available effect partially captured / info available, requires additional processing									
		Gain compression / expansion	Desensitisation	Intermodulation	Harmonics	Content at frequencies present in input		Content at frequencies not present in input	
		dependence of gain on input excitation level	dependence of response at one frequency on input at another frequency	combination of input frequencies to produce new frequencies in the output	generation of frequency components in output at multiples of input frequency	gain	phase	gain	phase
Conventional FRF									
Generalized FRF									
Nonlinear FRF									
Describing function		sinusoidal							
		generalized							
		HOSIDF							
Best linear approximation									

TABLE 1. Overview of frequency domain modeling approaches for nonlinear systems [23].

with the Arduino, and the user interacts with the controls through a web page.

CONCLUSION

Control techniques for precision mechatronic systems continue to develop with advances in available tools and evolving techniques. Feedforward techniques are gaining more attention despite the lack of emphasis on them in most undergraduate control systems classes. Feedback analysis remains grounded in classical frequency domain techniques, but these are evolving to address nonlinearities inherent in precision motion control systems, and to take advantage of the inexpensive access to experimental performance data. Finally, low-cost microcontrollers and sensors are enabling new teaching methods and distributed control with multiple processors where it was never before viable.

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Herds of Motion

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INTRODUCTION

In most manufacturing process, be it a high-end semiconductor manufacturing machine or low-end consumer electronics assembly lines, combined X and Y motion is always ubiquitously required to manipulate a workpiece to complete various manufacturing tasks. The traditional approach to XY motion stages is to stack multiple linear motion stages together. As a result, each motion stage claims a dedicated footprint without being able to share with other motion stages, and it is difficult to allow many motion stages to share a workspace.

A motion stage is generally an assistive component inside a machine, and the achieved motion is used to assist the implementation of some more important functionality of the machine. By looking from the point of view of machine functionality, it is found that stacked linear stages bring problems constraining manufacturing performance. In a machine, a part needs to be first installed on the motion stage with proper alignment/calibration, then the part is processed in the machine, and finally the part is uninstalled from the motion stage. Obviously, time is wasted during the above mentioned installation/uninstallation and productivity is compromised. In addition, the installation/uninstallation usually requires a dedicated robot, and thus imposes extra cost for machines. In a modern pipelined production process, a workpiece usually goes sequentially through a series of machines and each machine implements certain processing functionality on the workpiece. The installation/uninstallation associated time and cost happens at every step. Moreover, a workpiece transfer system is needed to carry workpieces from one machine to another.

A new paradigm is: a motion stage carries a workpiece to a machine, generates the planar motion required by the machine, then carries the workpiece out of the machine immediately after processing, and runs to another machine for

next step; at the same time another motion stage carrying a new workpiece rushes into the machine. The result will be: the machine can dedicate 100% of its time on the important functionality instead of idling and waiting for a robot to do repeated installation/uninstallation; robots are removed from every machine and cost is reduced; workpiece transfer system is eliminated as well.

Actually, if looking from further outside point of view, all manufacturing processes are expected to meet the needs of end-users/consumers. Motion stages should be designed in a way of considering how to improve for the overall value chain ending at customers. In the last century, manufacturing mean mass production; in this century, production soon means mass customization. Information technology infrastructure has linked manufacturers and end customers directly. Consequently, manufacturing process will be driven in real-time by the needs of individual customers each with a unique set of requirements. Ultimate flexibility in manufacturing will demand that each workpiece should be carried through a unique set of processing steps based on the end-user requirements.

All the above requirements call for a solution providing a large number of motion stages. Each motion stage is not only capable of required XY motion locally inside a machine, but also can convey workpieces among different machines in factory wide, from everywhere to anywhere at any time in any way that customers want. The benefit to the whole manufacturing process is improved productivity combined with reduced cost, and immediate response to personalized needs from customers. Figure 1 illustrates the concept of many concurrently-working motion stages in a next-generation manufacturing process: a herd of motion stages are busy inside a factory; each stage may be working inside a particular machine, may shuttle from one machine to another, and may queue up outside

a processing step. These functions are dynamically changed according to real-time customer requirements and optimized productivity and resource usage.

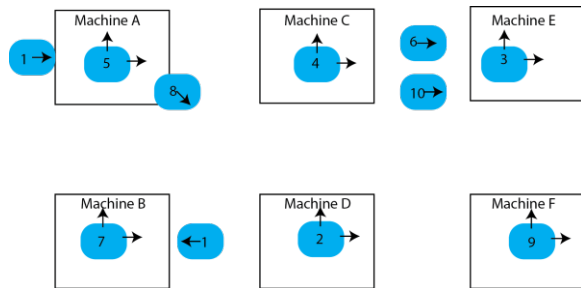


FIGURE 1. Herds of motion in a next-generation factory. Blue rectangles represent motion stages. Arrows indicate the traveling direction of motion stages.

FUNCTION REQUIREMENTS

There already exist planar motion stages, such as magnetically levitated stages used in ASML lithography machines, and Sawyer motors used in PCB drilling machines. However, these existing planar motion stages require an attached umbilical cord to supply electricity, and thus are limited to machine-wide motion, resulting in a very small number of concurrent working stages.

To technically bring planar motion stage 2.0 into reality, here we summarize a list of function requirements so that a general-purpose motion stage can be developed to meet the needs of next-generation factory.

- 1) Zero friction/lubrication/contamination. Friction brings many problems on positioning performance in addition to energy loss. Lubrication means extra cost for maintenance. Contamination is the enemy for high quality manufacturing. Therefore, these bad things should disappear in the next-gen factory.
- 2) High speed/acceleration/precision. These motion quality metrics are directly linked to productivity and yield. Four example, speed is expected to reach 5+m/sec and acceleration can be expected at 40+ m/sec². Precision should be able to tailor application requirements from μm to nm.

- 3) Non-stopped working. Motion stages should serve the manufacturing process 24/7 without lunch break.
- 4) Vacuum compatible. The technology should serve semiconductor industry that represents the highest requirements of modern manufacturing.
- 5) Wash-down compatible. The feature is required to make the developed technology to become truly general purpose.
- 6) Simple and easy operation.
- 7) Scalable to thousands of motion stages working in the range of hundreds of meters.
- 8) Low cost. The new solution must be cost-wise more effective/competitive than existing solutions (based on stacked linear stages).

OVERALL SOLUTION

As shown in Figure 2, the overall system comprises of stator coil assemblies and movers (only one is shown here). Each moveable stage comprises of a magnet array assembly to provide forces for driving and guiding the motion stages. There are four linear magnet arrays (X1, Y2, X3, Y4). In each array, there are a plurality of magnetization segments; each magnetization element is elongated along a particular linear direction. The moveable stage can travel on top of the stator coil assembly with a controllable air gap. The coil assembly comprises of multiple layers of coils carrying current that flows in two orthogonal directions (X, Y). The layers of coils extending in X direction are interlaced with those extending in Y direction. Such a coil assembly can be made from widely used printed-circuit board fabrication technology. Each magnet array interacts with its corresponding active coils (of the same color as the magnet array) to produce one levitation force (in Z) and one lateral driving force (in X or Y). Only coils close to the controlled motions stages are turned on (active) and are driven with controllable currents, and the rest of coils are switched off to maintain low power consumption. The combination of eight forces allows 6-DoF motion control of the moving magnet stage. The inactive coils are not shown in Figure 2. When the motion stage travels, the set of active coils are changed dynamically based on moveable stage positions. In each layer of the coil traces, the coil orientation is the same. The uniformity of coil distribution allows the creation of a travelling wave of current that is locked to the moving

stage. By properly commutating the current in each coil trace, the field generated by a group of active coils can be stationary relative to the permanent magnet array, and thus the generated force on the motion stage is position independent as the mover never experiences a change in the travelling wave of current as it traverses the working zone. As both the coils

and the magnet arrays are elongated in the same direction, the actuation force on the array is naturally insensitive to the linear motion into the page. When the stage travels laterally, the group of active coils will shift with the stage. For vertical motion, the current amplitude of each coil will be adjusted based on the exponential decay of the field with vertical position.

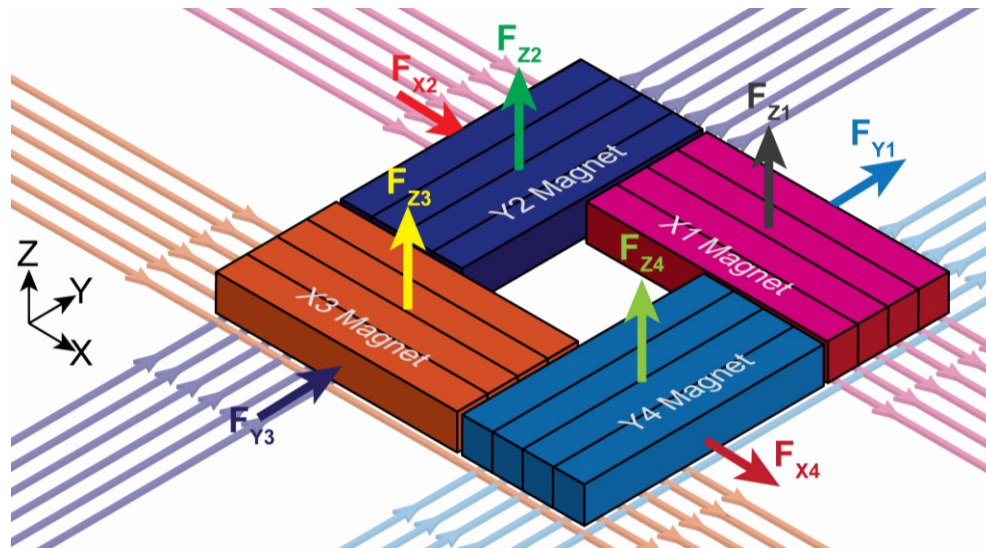


FIGURE 2. Overall system configuration.

COIL CURRENT COMMUTATION

Here an example is used to illustrate coil commutation process. Figure 3a shows a Y-elongated magnet array (such as Y2 or Y4) hovering above a set of coils linearly elongated in the Y direction. The magnet array M_0 has an X-direction width λ (one spatial period of magnet array magnetization direction). The magnetic field generated by the magnet array is shown as B_{z0} in red. Obviously, such a magnetic field has no periodic pattern as commonly seen in conventional linear motors. Therefore a current-commutation law is not obvious. However, if checking the force between the magnet array M_0 and a group of coils ($C_0, C_1, C_{-1}, \dots$), each of which is spaced from one another by the magnet array width and is excited by the same current, it is equivalent to the force between single coil C_0 and a group of magnet arrays ($M_0, M_1, M_{-1}, \dots$) as shown in Figure 3b, each of which is spaced from one another by the magnet array width. The coil-magnet array force in Figure 3b is exactly what is inside a conventional linear motor: a coil is immersed inside an infinitely extended periodic magnet array as shown in Figure 3c. As a result, the sinusoidal

commutation law conventional linear motors can be directly applied to the case of narrow magnet array interacting with repeating coils in Figure 3a. In reality, only coils extending within $\lambda/2$ from magnet array edges needs to be excited, as others contribute nearly zero forces even if they are turned on. It should be noted that the equivalence discussed above is under the matching condition among magnet array width, its magnetization spatial period, and coil spatial period.

M-MAGNET ARRAY

Although Halbach magnet arrays (made of both horizontally magnetized segments and vertically magnetized segment) are advantageous in terms of lower harmonic field and flux concentration at the working side, it is difficult to have symmetric design and a width of integer number of its spatial period. Symmetric design is important to produce zero force/torque coupling between X and Y. In the Halbach magnet array in Figure 3a, the width of magnetization segments near the edges is half of inner segments away from the edges, in order to produce a symmetric field. A recently

developed new magnet array is shown in Figure 4: it is made of single type of magnetization segments 45-degree magnetized segments. Since the array magnetization pattern is in M shape, they are called M-magnet array. In addition to having all advantageous features that previous symmetric Halbach arrays can offer, the new design is made of fewer segments and fewer segment types, and is much easier to assemble at lower cost.

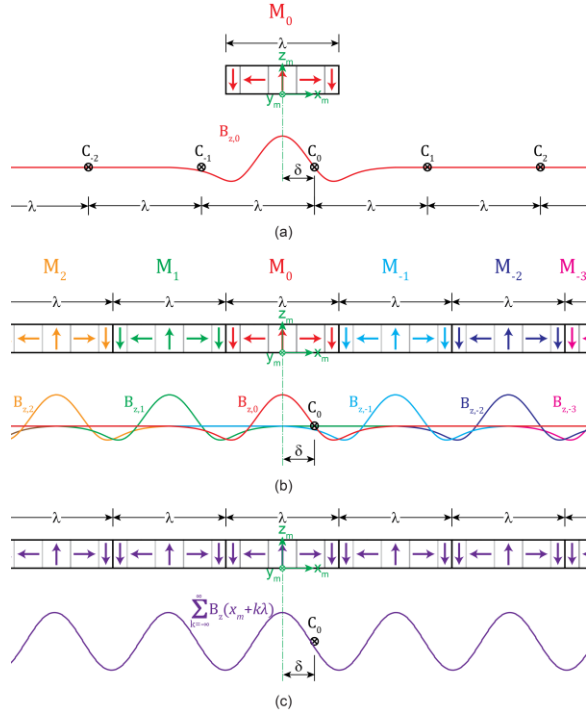


FIGURE 3. Illustration of coil current commutation.

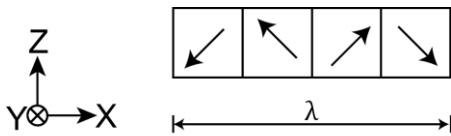


FIGURE 4. M-magnet array.

MODULAR STATOR

The whole stator is made up of modular tiles, as illustrated in Figure 5. This configuration allows a system to be arbitrarily expanded. Each stator tile is made up of PCB assemblies dedicated for motion, power, and information processing, respectively. All tiles are connected to both a common DC power bus and a high speed network for data exchange.

SUMMARY

Basically each motion stage can be controlled at high bandwidth such as 100 Hz bandwidth for all

6 axis, thanks to simple mechanical structure. The RMS positioning noise can be less than 1 micron for X/Y/Z linear axis and dozens of micro-rad for Rx/Ry/Rz rotational axis. In conclusion, supporting technologies for herds of motion are readily available for next-generation factory.

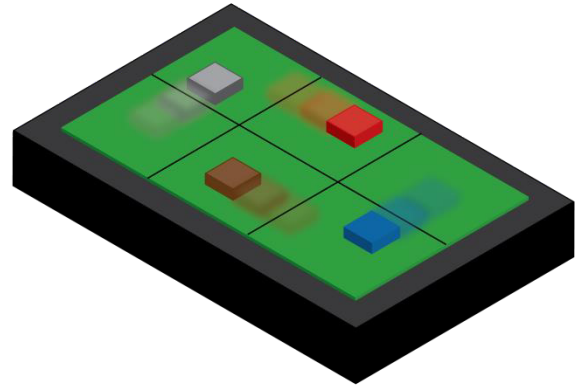


FIGURE 5. Planar motion stages with multiple stages.

NON-COLLOCATED OPTIMAL CONTROL OF FLEXIBLE DRIVES WITH LOAD SIDE FEEDBACK

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INTRODUCTION

This paper presents a novel optimal motion control technique for feed drive systems with structural flexibility. The structural resonance is modeled and actively compensated within the controller structure utilizing only the load side feedback, i.e. in a non-collocated fashion. Controller gains are tuned to inject modal damping and attain high positioning bandwidth with vibration suppression. Mainstream industrial controllers rely on the convenient P-PI cascade control structure to achieve moderate positioning performance [1]. However, structural flexibilities limit their performance. For instance, high gain P-PI controllers implemented with motor side feedback (collocated operation) may cause severe vibrations on the load side. On the other hand, if they are implemented with load side feedback (non-collocated operation) they suffer from instability. As a result, cascade P-PI controller structure is not a viable solution for accurate, high-bandwidth control of high speed feed drive systems [2], [3]. In order to cope with that, this work utilizes load side position, velocity, acceleration and jerk measurements to control the structural dynamics of the flexible drive. The pole locations are assigned optimally based on the Linear Quadratic Regulator (LQR) framework by weighting modal damping and tracking objectives [5].

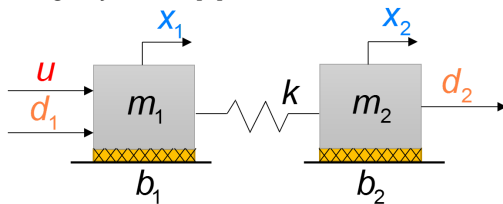


FIGURE 1. Free body diagram of a typical flexible feed drive system.

SYSTEM MODELING AND CONTROL LAW

Typically, a flexible feed drive system can be modeled as a 2DOF system based on the free body diagram shown in Fig .1. The governing set of differential equations can be derived as

$$\begin{aligned} m_1 \ddot{x}_1 &= -b_1 \dot{x}_1 + k(x_2 - x_1) + u \\ m_2 \ddot{x}_2 &= -b_2 \dot{x}_2 + k(x_1 - x_2) \end{aligned} \quad (1)$$

where x_1 and x_2 are displacements of the masses m_1 and m_2 . m_1 denotes the mass of the motor and m_2 denotes mass of the load/table. u is the motor force, or so-called the equivalent control input that drives the system. b_1 and b_2 are viscous friction forces and k is the equivalent spring constant representing the flexibility in-between masses, i.e. motor and table. Typically, flexibilities in a feed drive can be originated from any structural element in the feed drive system. For instance, the screw-nut interface introduces a lightly damped flexibility in most ball-screw driven motion systems [3].

Equation (1) can be re-written to express the dynamics of the table position x_2 with respect to the control input, u directly as:

$$\frac{k}{m_1 m_2} u = x_2^{(iv)} + a_3 \ddot{x}_2 + a_2 \dot{x}_2 + a_1 x_2 \quad (2)$$

where

$$\begin{aligned} a_1 &= \frac{(b_1 + b_2)k}{m_1 m_2}, \quad a_2 = \frac{b_1 b_2 + (m_1 + m_2)k}{m_1 m_2}, \\ a_3 &= \frac{b_1 m_2 + b_2 m_1}{m_1 m_2} \end{aligned} \quad (3).$$

The 4th order differential equation presented in (2) can be re-written as a set of 1st order differential equations, i.e. in state space form, by selecting the state vector as higher order derivatives of the table position

$\mathbf{v} = [x_2 \ \dot{x}_2 \ \ddot{x}_2 \ \ddot{x}_2^{(iv)}]^T$ as follows

$$\begin{bmatrix} \dot{x}_2 \\ \ddot{x}_2 \\ \ddot{x}_2 \\ x_2^{(iv)} \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & -a_1 & -a_2 & -a_3 \end{bmatrix}}_{\mathbf{A}} \underbrace{\begin{bmatrix} x_2 \\ \dot{x}_2 \\ \ddot{x}_2 \\ \ddot{x}_2 \end{bmatrix}}_{\mathbf{v}} + \underbrace{\begin{bmatrix} 0 \\ 0 \\ 0 \\ k/m_1 m_2 \end{bmatrix}}_{\mathbf{\beta}} u \quad (4)$$

Note that the representation in (4) is in controllable canonical form and the eigenvalues of the system can be computed as the roots of the 4th order polynomial,

$$s(s^3 + a_3s^2 + a_2s + a_1) = 0 \quad \text{or} \quad (5)$$

$$s(s + \lambda)(s^2 + 2\zeta\omega_n s + \omega_n^2) = 0$$

which are the eigenvalues of $\mathbf{\Lambda}$,

$$\text{eig}(\mathbf{\Lambda}) = \begin{cases} p_1 = 0 \\ p_2 = -\lambda \\ p_{3,4} = -\zeta\omega_n \pm j\omega_n\sqrt{1-\zeta^2} \end{cases} \quad (6).$$

The 4 eigenvalues in (6) govern the rigid body and the vibratory dynamics. p_1 is the integral action that converts table velocity to table position, p_2 (or $-\lambda$) is the rigid body pole and p_3 and p_4 are the complex conjugate vibratory poles with the damping ratio ζ and natural frequency ω_n .

Furthermore, based on the state space representation given in (4), a pole placement controller can be designed to locate all the poles at desired locations and realize direct feedback control by simply measuring the load side position x_2 , velocity $\dot{x}_2$, acceleration $\ddot{x}_2$ and jerk $\ddot{\ddot{x}}_2$. In other words, a state feedback control law can be written as

$$u = \mathbf{K}_{1 \times 4}(\mathbf{v}_r - \mathbf{v}) \quad (7)$$

$$= k_p(x_{2r} - x_2) + k_v(\dot{x}_{2r} - \dot{x}_2) + k_a(\ddot{x}_{2r} - \ddot{x}_2) + k_j(\ddot{\ddot{x}}_{2r} - \ddot{\ddot{x}}_2)$$

where $\mathbf{v}_r = [x_{2r} \ \dot{x}_{2r} \ \ddot{x}_{2r} \ \ddot{\ddot{x}}_{2r}]^T$ is the reference state vector containing higher order derivatives of the trajectory, and $\mathbf{K} = [k_p, k_v, k_a, k_j]$ contains feedback gains, which can be computed based on the desired dynamics using conventional pole assignment techniques such as Ackermann's method [4].

Thus, by measuring the table position, velocity, acceleration and jerk dynamics of feed drive can be designed freely. In other words, the positioning bandwidth as well as damping of the vibration can be controlled [6]. However, such a controller is inherently not robust against disturbances and unmodelled dynamics. This can be tackled by introducing integral action to the feedback controller. Original state space representation from (4) is augmented with an integral state,

$$\begin{bmatrix} \dot{\mathbf{v}} \\ \dot{x}_2 \end{bmatrix} = \underbrace{\begin{bmatrix} \mathbf{\Lambda} & \mathbf{0}_{4 \times 1} \\ 1 & \mathbf{0}_{1 \times 4} \end{bmatrix}}_{\mathbf{A}} \underbrace{\begin{bmatrix} \mathbf{v} \\ \int_0^t x_2(\tau) d\tau \end{bmatrix}}_{\mathbf{x}} + \underbrace{\begin{bmatrix} \boldsymbol{\beta} \\ 0 \end{bmatrix}}_{\mathbf{B}} u \quad (8)$$

and the control law is updated as,

$$u = \mathbf{K}_{1 \times 5}(\mathbf{x}_{r2} - \mathbf{x}) \quad (9).$$

$$= k_p(x_{2r} - x_2) + k_v(\dot{x}_{2r} - \dot{x}_2) + k_a(\ddot{x}_{2r} - \ddot{x}_2) + k_j(\ddot{\ddot{x}}_{2r} - \ddot{\ddot{x}}_2) + k_i \int_0^t (x_{2r}(\tau) - x_2(\tau)) d\tau$$

As observed, above control law can be implemented as a PID action on the position feedback with a parallel PD loop operating acceleration feedback in industrial controllers.

OPTIMAL POLE ASSIGNMENT

The basic pole placement controller presented in the previous section can be designed by carefully assigning all the 5 closed loop poles associated with the rigid body and vibratory dynamics. Typically, closed loop poles of flexible feed drive systems are determined either based on heuristics, approximations [7] or through engineer's expertise. The goal in tuning most industrial feed drive systems is to achieve good tracking and disturbance rejection and the same time dampen structural vibrations for precision positioning. Although these objectives are not directly contradicting with each other, significant control action is necessary to achieve both of them, which in return demands large actuation forces and may cause saturation. The objective in this section is to determine optimal closed loop pole locations to balance the feed drive performance for tracking, vibration suppression and control effort.

An optimal control strategy that seeks a balance between these objectives can be designed using the Linear Quadratic Regulator (LQR) framework [5]. The problem is to find the optimal pole locations which minimizes the following performance index

$$J = \int_0^\infty (\mathbf{x}^T \mathbf{Q} \mathbf{x} + \mathbf{u}^T \mathbf{R} \mathbf{u}) dt \quad (10).$$

The first term in (10) penalizes deviation of the states from a set point and second term penalizes the control action. Here, $\mathbf{Q}$ is the constant weight matrix, which increases contribution of each state, and $\mathbf{R}$ weights the control effort. Notice that since the pole placement problem given in (4) is single input multiple output, $\mathbf{R}$ becomes a scalar gain. However, design of the $\mathbf{Q}$ matrix is not trivial to achieve tracking and vibration suppression at the same time. The following analysis is undertaken to design the state weight matrix $\mathbf{Q}$.

The first objective is to decide on which state must be penalized in the performance index to suppress structural vibrations.

Firstly, assuming that viscous friction is small in precision feed drive systems, $b_1=b_2=0$, (2) is simplified as follows

$$x_2^{(iv)} + \underbrace{\frac{k(m_1 + m_2)}{m_1 m_2}}_{\omega_n^2} \ddot{x}_2 = \frac{k}{m_1 m_2} u \quad (11).$$

which clearly shows dynamics of the undamped vibratory mode. Next, the following control law based on sole jerk feedback ($\ddot{x}_2$) is postulated to inject damping into the system

$$u = -\phi \ddot{x}_2, \quad \phi \in \mathbb{R} \quad (12).$$

Substituting (12) in (11) yields

$$x_2^{(iv)} + \phi \underbrace{\frac{k}{m_1 m_2}}_{2\zeta\omega_n} \ddot{x}_2 + \underbrace{\frac{k(m_1 + m_2)}{m_1 m_2}}_{\omega_n^2} \ddot{x}_2 = 0 \quad (13).$$

As shown in (13), feeding back table jerk ($\ddot{x}_2$) directly influences vibratory dynamics by injecting damping. In other words, minimizing deviation of the jerk of the load from the set-point acts as dampening effect on the structural vibrations, and linearly increases damping of the resonant mode, e.g. $\zeta^{new} = \phi \zeta^{old}$. This indicates that the performance index in (10) should penalize the jerk state of the table for robust vibration damping.

As observed from (13), minimizing jerk of the table does not affect velocity dynamics (i.e. rigid body dynamics) of the drive system. In other words, static compliance or the tracking bandwidth of the feed drive system are not altered. Nevertheless, tracking and disturbance rejection are the other performance pillars in motion control. Particularly, integral of the load

side position ($\int_0^t x_2(\tau) d\tau$) is an indicator for

tracking and low frequency disturbance rejection performance. Thus, the performance index in (14) should also generate integral action, and hence by defining $\mathbf{H} = [0 \ 0 \ 0 \ \alpha \ 1]$ both jerk and integral of the table are penalized in (10) as:

$$J = \int_0^\infty (\rho \mathbf{x}^T \underbrace{(\mathbf{H}^T \mathbf{H})}_{\mathbf{Q}} \mathbf{x} + u) dt \quad (14).$$

$$= \int_0^\infty (\rho \left(\underbrace{\alpha \ddot{x}_2 + \int_0^t x_2(\tau) d\tau}_{\mathbf{z}} \right)^2 + u) dt$$

Notice that $\mathbf{z}$ is a state composed of weighted linear combination of table jerk and integral states. This formulation allows us to represent the system as a SISO form where u is the input and $\mathbf{z}$ is the output. α is a scalar weight and becomes a tuning parameter to balance between vibration suppression and tracking. ρ is the second tuning parameter of the proposed controller, which weights control action, i.e. control effort, over control performance.

Finally, the optimal state feedback gains $\mathbf{K}_{1 \times 5}$ that minimizes the performance index J is obtained by solving the Algebraic Riccati Equation defined by $\mathbf{A}$, $\mathbf{B}$, ρ and α in the LQR framework [5], and implemented in the form of (9) as a $\text{PID}_{pos} + \text{PD}_{acc}$ type feedback controller.

EXPERIMENTAL SETUP AND CONTROLLER PERFORMANCE

Previous section presented the proposed optimal motion controller scheme. This section validates performance and practicality of the controller experimentally. The experimental 2DOF linear motor driven drive system is shown in Fig. 2. The cart on the left hand side is driven by a linear motor and connected to a load on the right hand side with a pair of linear springs. Both carts are equipped with resolvers with 4 [μm] resolution. The servo amplifier is set to operate on torque control and the control signal is generated at a real-time computer at 1 kHz.

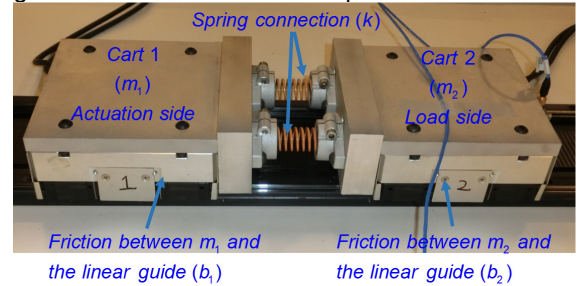


FIGURE 2. Experimental Setup.

Firstly, system dynamics is identified through transfer function fitting in frequency domain. Fig. 3 shows actual measured frequency responses from both carts, namely x_1 and x_2 . Identified transfer functions between control signal to cart positions are:

$$\frac{X_1(s)}{U(s)} = \frac{2671(s^2 + 7.881s + 6791)}{s(s + 2.752)(s^2 + 9.872s + 13700)} \quad (15)$$

$$\frac{X_2(s)}{U(s)} = \frac{1.875(10^7)}{s(s + 2.752)(s^2 + 9.872s + 13700)} \quad (16)$$

Due to rough resolution of resolvers, an accelerometer with a sensitivity of 102.0 mV/m/s² is attached to the second cart, and it is used to measure acceleration and jerk states. In the actual implementation, a reduced order observer is designed to realize high bandwidth acceleration and jerk state measurements.

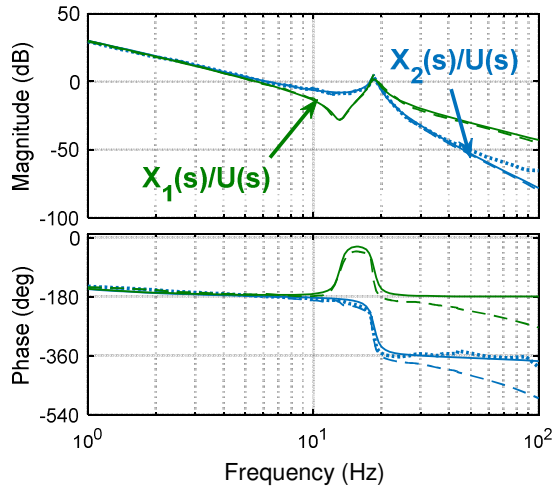


FIGURE 3. FRFs for ideal models (solid lines), encoder measurement (dashed lines) and accelerometer measurement (dotted line, only for 2nd cart) regarding the experimental setup.

As observed from Fig. 3, the system has a vibratory mode at 117 rad/s (18.62 Hz) with 4.2% damping and a rigid body mode at 2.752 rad/s (0.438 Hz). Firstly, parameters of the proposed controller are tuned for moderate control performance as $\alpha=0.292(10^{-3})$ and $\rho=3(10^3)$. The rationale behind the selection of α and ρ will be discussed in the following section. Proposed controller is benchmarked against the conventional P-PI controller with motor side velocity and table side position feedback loops. The P-PI control law 3 tunable gains yields;

$$u_{PPI} = K_V [(x_{2r} - x_2)K_P + \dot{x}_{2r} - \dot{x}_1] + K_I \int_0^t [(x_{2r}(\tau) - x_2(\tau))K_P + \dot{x}_{2r}(\tau) - \dot{x}_1(\tau)] d\tau \quad (17)$$

K_V and K_I are proportional and integral velocity gains and K_P is the table-side position feedback gain. These 3 parameters only allow placement of 3 closed loop poles. As opposed, proposed controller can place 5 of the closed loop poles freely. In order to compare the P-PI controller performance to the proposed controller fairly, 2 of these poles are chosen to match velocity dynamics poles of the proposed controller. The 3rd pole is chosen to be same as the rigid body

pole of the open loop system and compensated by the PI velocity loop. The remaining 2 poles became residuals that are self-assigned. Such an arrangement yields comparable gain cross-over frequencies and phase margins for the experimental setup. Table 1 summarizes the pole locations, controller parameters and frequency domain characteristics of P-PI and proposed optimal controller.

TABLE 1. Closed loop poles, controller parameters and frequency domain characteristics of proposed and P-PI controllers.

	Proposed		PPI	
Closed loop poles [rad/s]	$-23.3 \pm 32.7i$		$-23.3 \pm 32.7i$	
	$-45.7 \pm 110i$		$-23.3 \pm 101i$ (residual)	
	-44.4		-2.752	
Controller Parameters	k_p	3.1635	K_V	0.0298
	k_v	0.089		
	k_a	$6.72(10^{-4})$	K_I	0.082
	k_j	$9.55(10^{-6})$		
	k_i	54.7723	K_P	29.6242
Gain cross-over freq. [Hz]	28.50		23.87	
Phase Margin [deg.]	41.6		45.7	

Fig. 4 represents loop transmission and the Nyquist plots for the controllers. Notice that while proposed controller has slightly higher cross-over frequency, P-PI controller shows a slightly larger phase margin. Fig. 5 shows experimental tracking performance comparison on a jerk limited point-to-point trajectory. As shown, proposed controller tracks the reference trajectory more accurately and with less vibrations. Fig. 6 gives more insight on the controller. As observed from the tracking and load side disturbance frequency responses, proposed controller shows much larger tracking bandwidth and much greater disturbance rejection capabilities. Moreover, the residual closed loop poles of the P-PI controller are slower than the damped oscillatory poles of the proposed controller. This lowers the disturbance performance of P-PI controller.

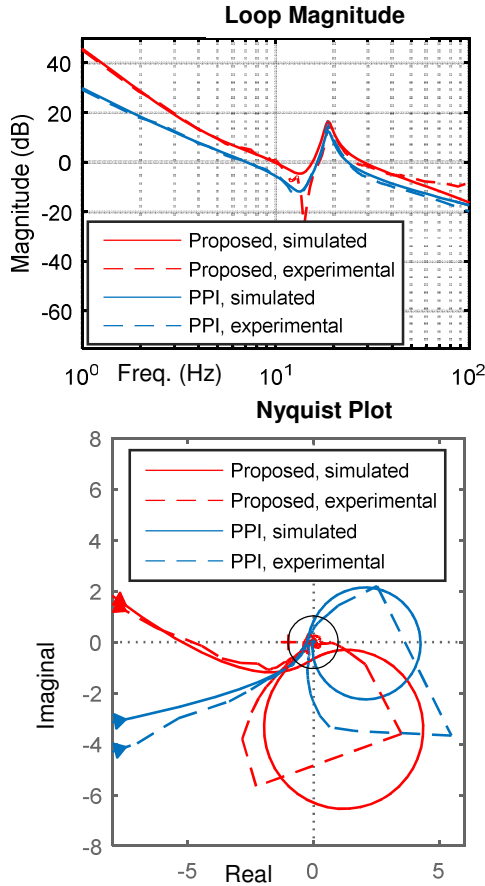


FIGURE 4. Loop transmission characteristics for both controllers in frequency domain.

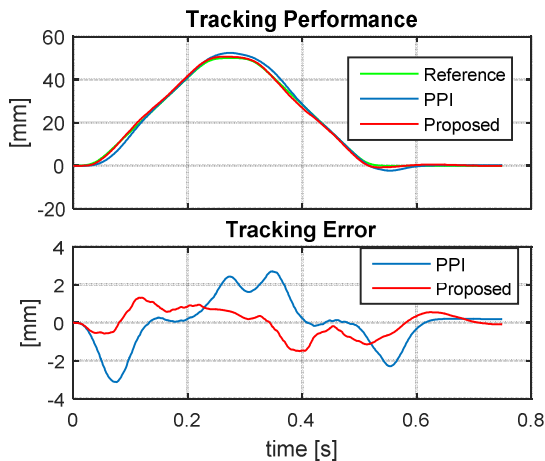


FIGURE 5. Tracking performance of proposed and P-Pi controllers.

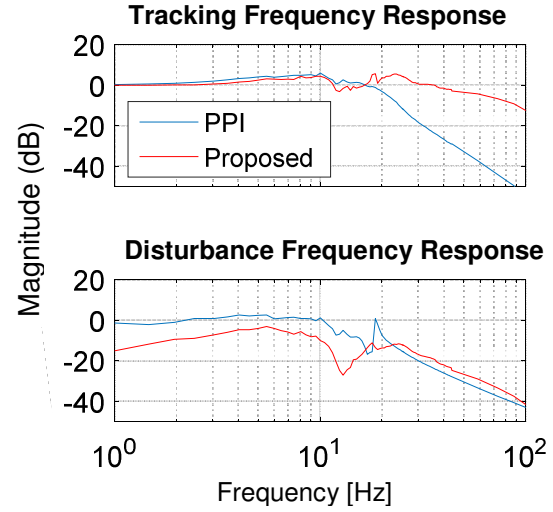


FIGURE 6. Tracking and load disturbance rejection for proposed and PPI controllers.

CONTROLLER ANALYSIS

Proposed controller has only two tuning parameters, namely α and ρ . This section analyses effect of controller parameters and presents tuning guidelines. In the proposed controller, α gain weights damping over tracking, and ρ weights the control action over control performance. The trajectory of the closed loop pole locations can be obtained using the symmetric root locus technique [5] by adding right hand side complementary poles ($-s$) to the closed loop system as

$$\Pi(s) = 1 + \rho \frac{[H(-sI - A)^{-1}B]^T [H(sI - A)^{-1}B]}{G^*(s)} \quad (18)$$

where $G^*(s)$ is the open loop transfer function with mirrored poles and additional zeros controlled by α . Considering only left hand side poles, (18) presents optimal closed loop pole locations with varying ρ . $G^*(s)$ can be represented based on (16)

$$G^*(s) = \rho \underbrace{(s^8 \alpha^2 + 2s^4 \alpha + 1)}_{N(s)} \frac{X_2(-s)}{U(-s)} \frac{X_2(s)}{U(s)} \quad (19)$$

which shows that G^* has 8 zeros (4 repeated zero couples) that are controlled by α gain as

$$s_m = \frac{1}{\sqrt[4]{\alpha}} e^{-j \frac{(2m-1)\pi}{4}}, \quad -1 \leq m \leq 2, \quad m \in \mathbb{Z} \quad (20).$$

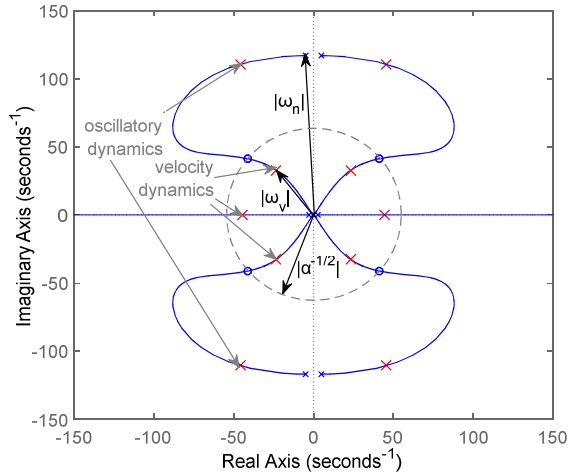


FIGURE 7. Symmetric root locus and optimal closed loop poles (shown in red, on the left half plane).

Fig. 7 illustrates symmetric root locus of the system based on (18). Optimal closed loop poles are the ones that are on the left half side of the complex plane. The trajectories of the closed loop poles are controlled by α . In other words, since increasing ρ simply moves closed loop poles to the zeros of G^* , bandwidth of the system is guided by α . α should be selected moderately larger than the desired natural frequency of underdamped closed loop rigid body poles, ω_v . Notice that, the final tracking bandwidth of the system will be larger than ω_v due to feed-forward terms in (9). On the other hand, assuming that the resonant mode frequency ω_n is higher than ω_v , α should be selected as

$$\omega_n > \frac{1}{\sqrt[2]{\alpha}} > \omega_v \quad (21)$$

Based on the analysis, following guideline is proposed for effective tuning of the controllers. Firstly, natural frequency of underdamped closed loop rigid body poles, ω_v , is selected as one third of the resonant frequency of the feed drive system as $\omega_v = \frac{\omega_n}{3}$. Then, α is tuned to place zeros of G^* at the half of the resonant mode $\frac{1}{\sqrt[2]{\alpha}} = \frac{\omega_n}{2}$. Finally, ρ is increased until the desired value of ω_v is reached on the symmetric root locus.

CONCLUSION

This paper has presented a high order non-collocated closed loop controller for flexible feed drives. Proposed controller is formulized as a state feedback pole placement controller with an additional integral state using only load side information. The decision for closed loop pole locations are made by defining an optimal control strategy that takes control effort, tracking, disturbance rejection and vibration suppression performances into account. The resulting pole placement strategy reduces the decision making process for 5 closed loop poles into tuning 2 optimal control parameters. Experimental results show that the designed controller outperforms industrial P-Pi controllers with a single feedback sensor and the tuning is significantly intuitive.

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FAST SCANNING OF X-RAY OPTICS: AN OPTIMAL CONTROL APPROACH

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INTRODUCTION

The Advanced Photon Sources Upgrade (APS-U) is one of a new generation of synchrotron X-ray light sources [1]. The APS-U will be 100 times brighter than the current APS. This increase in brightness will reduce scanning times and increase the optical resolution [2]. In fact, to use all of the available photons and not damage samples, the X-ray beam will need to be scanned much more rapidly across the sample. This emphasis of fast scanning and high optical resolution makes rapid, stable control of the optics scanning stage paramount.

The scope of this paper is to highlight the performance enhancing controller algorithms that would make it possible to achieve large positioning bandwidth, good positioning resolution, robustness to unmodeled uncertainties, sufficient environmental disturbance rejection and adequate noise attenuation. An optics scanner capable of such performance would be able move along any target trajectory with less than 1% tracking error within a predefined closed-loop bandwidth. This would also counter the effects of thermal drift of the scanning stages.

THE VELOCIPROBE X-RAY MICROSCOPE

A new fast-scanning microscope is under construction at the APS. The Velociprobe is designed to scan a $1\text{ }\mu\text{m}$ by $1\text{ }\mu\text{m}$ area in 10 s or less at a resolution of 10 nm. The Velociprobe comprises a custom-designed, stable coarse positioning stages and state-of-the-art Physik Instrumente (PI) piezo nanopositioning stages. The high-precision and real-time displacement of the PI XYZ optics nanopositioning stages will be measured by an Attocube FPS-3010 interferometric sensor system. The Attocube sensor uses fiber-guided lasers reflected from small mirrors mounted close to the sample

position. The returning interference signal in the optical fiber is then processed in the Attocube electronic unit to provide a quadrature signal, which is in turn processed on-the-run to achieve the displacement signal in real-time. In case of Velociprobe, six separate quadrature signal will be read by the digital I/O modules of National Instruments (NI) cRIO hardware and processed in the built-in FPGA in the cRIO chassis. The reference trajectory for scanning is given as a voltage signal from NI Analog Output modules to the PI amplifier for the respective stages.

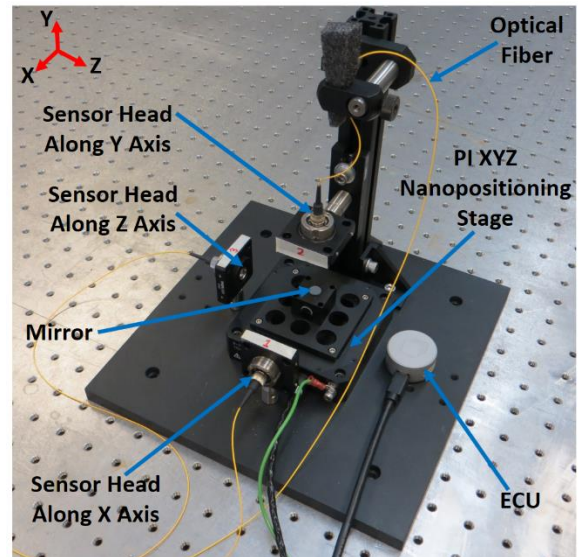


FIGURE 1. The fine-scanning stages of the Velociprobe X-Ray microscope comprises of PI P-733.3DD XYZ piezo nanopositioner. Three Attocube sensor heads are positioned in the direction of three mirrors glued on a cubic mount that is attached on the PI stage. The Attocube Environmental Compensation Unit (ECU) measures air temperature, pressure, and humidity to correct for the variation of index of refraction due to changing properties of air.

IDENTIFICATION

The PI XYZ stages are of a proprietary parallel kinematics design. The unknown geometry and complex design make developing a model from first principals impossible. Here, we are using a frequency response based experimental identification technique to generate a mathematical model for the nanopositioning stages. A bandlimited uniform white noise was given as an input to each of the nanopositioning stages and the corresponding displacement were measured using the high precision Attocube interferometric sensors. Frequency response is then calculated from these time-domain I/O data for each of the 3 diagonal (XX, YY, and ZZ) and 6 off-diagonal stages (XY, YX, YZ, ZY, ZX, and XZ). A parametric transfer function is then fit to the frequency response data.

To account for the nonlinearities present in the stage dynamics, variation in transfer function models at different operating points was studied. A linear plant would behave identically at different operating points. If the variation of the plant dynamics is small in the region of interest, then we can expect a linear model to be sufficient for the controller design. Accordingly, to model the stage dynamics, we took the average of the plant dynamics obtained at nine separate operating points distributed over the volume of operation of the XYZ stages.

X-Stage Identification

The identification of the diagonal XX-stage (model identified from the X-stage Input in mV and X-stage displacement in nm) and off-diagonal XY and XZ stages are detailed in this section.

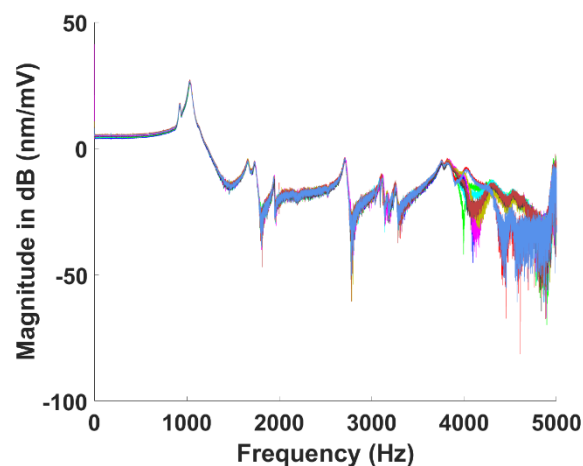


FIGURE 2. The transfer function estimates at 9 separate operating point for XX-stage. The stage

dynamics shows variation with the changing operating point in the frequencies higher than 3800 Hz.

The variation of XX-stage dynamics with changing operating point is shown in Figure 2. The 9 different XX-stage dynamics is then averaged to obtain the average dynamics (Fig 3).

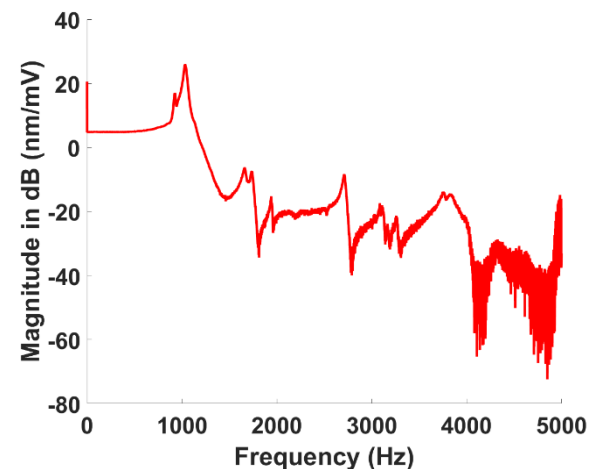


FIGURE 3. This figure shows the average transfer function estimate for XX-stage. The first resonance peak is at 1034 Hz.

Similarly, the variation of XY-stage dynamics and XZ-stage dynamics at same 9 operating points are shown in Figure 4 and Figure 6 respectively. While XY-stage dynamics shows minimal variation, the XZ -stage dynamics do show some differences in the high frequency region. Figure 5 and Figure 7 respectively shows the average dynamics XY and XZ stages.

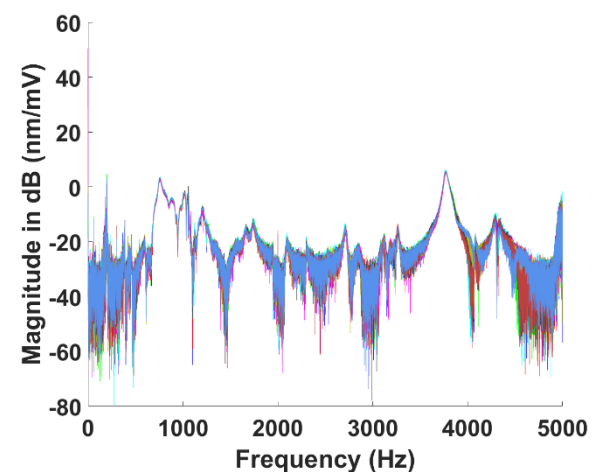


FIGURE 4. This figure shows the transfer function estimates at 9 separate operating point for XY-stage with minimal variation being present.

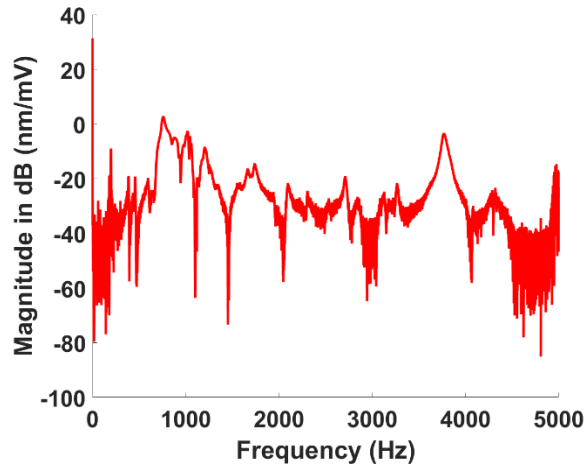


FIGURE 5. The average transfer function estimate for XY-stage is shown with the first resonance peak at 757 Hz.

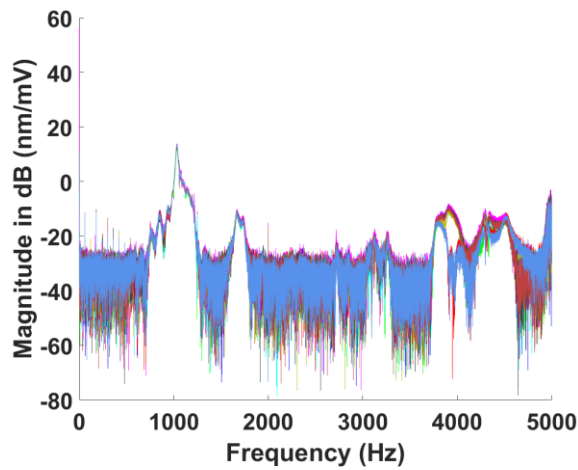


FIGURE 6. The transfer function estimates at 9 separate operating point for XZ-stage is shown. The stage dynamics varies with the changing operating point in the high frequency region only.

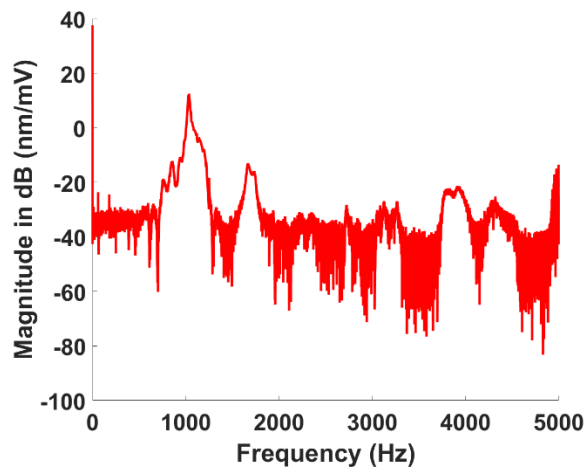


FIGURE 7. This figure shows the average transfer function estimate for XZ-stage. The first resonance peak is at 1033 Hz.

Y-Stage Identification

In this section the identification of the vertical stage Y (diagonal YY-stage and off-diagonal YX and YZ stage) is explained. The variation of YX, YY, and YZ stage dynamics at operating points distributed around the operating volume are observed similarly to that of X-stage.

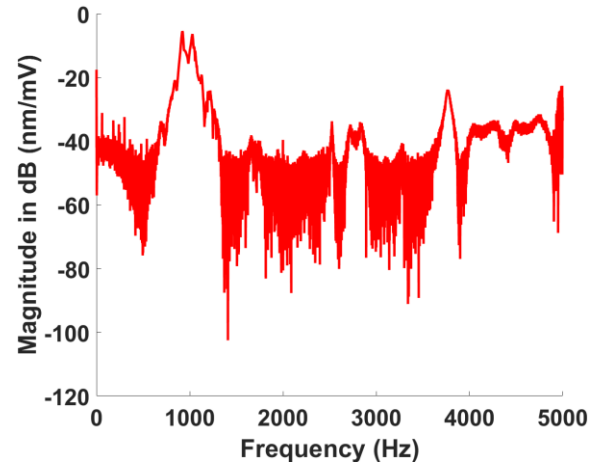


FIGURE 8. The average transfer function estimate for the off-diagonal YX-stage is shown above. The first resonance peak is at 923 Hz.

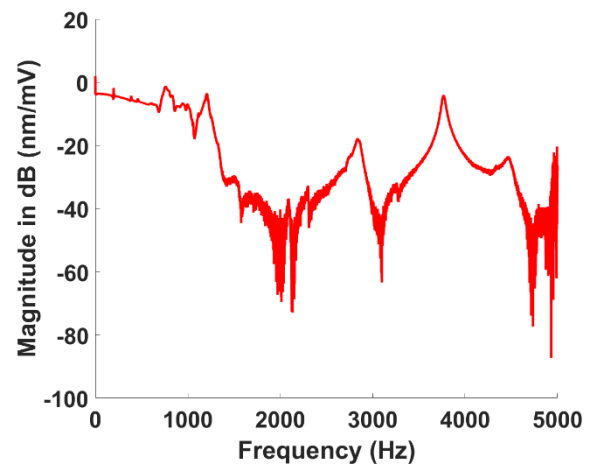


FIGURE 9. This figure shows the average transfer function estimate for the diagonal YY-stage. The first resonance peak is at 758 Hz.

The averaged YX, YY, and YZ stage dynamics are shown in the Figure 8, Figure 9 and Figure 10 respectively.

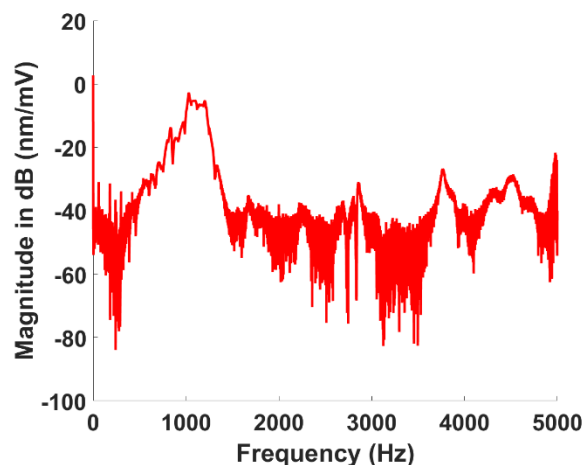


FIGURE 10. The average transfer function estimate for YZ-stage is shown with the first resonance peak is at 836 Hz.

Z-Stage Identification

Similar studies on the variation of the ZX, ZY, and ZZ stage dynamics with changing operating point around the volume of stage operation was carried on but shown here.

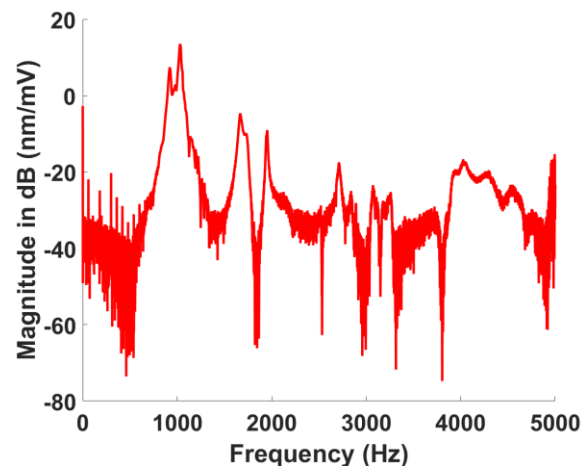


FIGURE 11. The average transfer function estimate for ZX-stage are seen here with the first resonance peak at 920 Hz. The twin peaks seen here corresponds to the resonance peaks of XX-stage.

The averaged ZX, ZY, and ZZ stage dynamics are shown in the Figure 11, Figure 12, and Figure 13, respectively. The diagonal ZZ-stage has a first resonance peak just above 1 kHz. The resonance peaks of the off-diagonal ZX and ZY stages corresponded to the resonance peaks of diagonal XX and YY stages respectively.

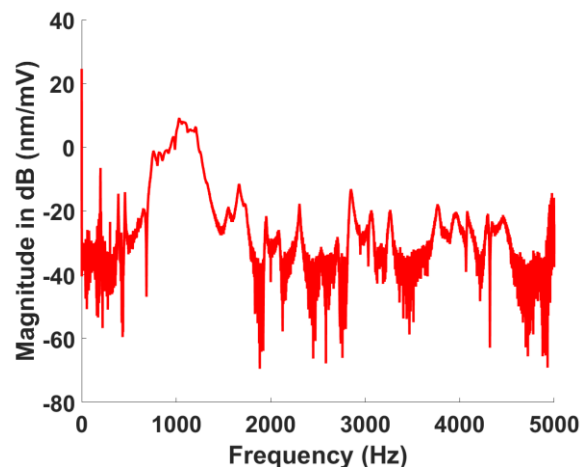


FIGURE 12. This figure shows the average transfer function estimate for ZY-stage. The first resonance peak is at 760 Hz.

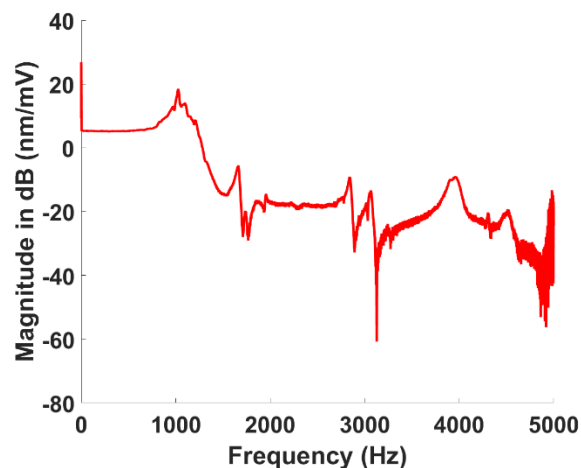


FIGURE 13. This figure shows the average transfer function estimate for ZZ-stage with the first resonance peak at 1026 Hz.

Curve Fitting

Controller designs only for the diagonal single-input single-output (SISO) XX-stage will be presented in this abstract. A parametric transfer function model G_{xx} is fitted to the average transfer function estimate calculation for the diagonal XX-stage. The frequency response of the 18th order model G_{xx} and the transfer function estimate calculation obtained from experimental I/O data is shown in Figure 14. This 18th order SISO model G_{xx} is used in later sections for designing the advanced controllers. The model for the remaining two diagonal stages YY and ZZ were done similarly. In case of multi-input multi-output (MIMO) controller design, a MIMO plant fitted to the 9 transfer function estimate calculations will be used, but is not shown here.

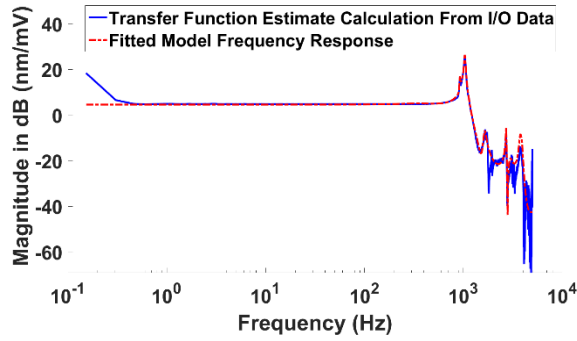


FIGURE 14. This figure shows the comparison between the transfer function estimate (blue solid line) of XX-stage calculated from XX-stage time-domain I/O data and the frequency response of the model G_{xx} (red dashed line).

CONTROL DESIGN

The identified stage models are being used to design control algorithms. H_∞ optimal control algorithm for the diagonal stages and MIMO stages will be designed and implemented. The target is to minimize the tracking error $r - y$ and reduce it to less than 1% up to a certain prespecified bandwidth in the closed-loop.

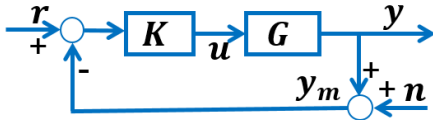


FIGURE 15. This figure shows transfer function block diagram of closed-loop system in negative feedback format. G is the identified plant model, K is the controller transfer function, r is the reference trajectory, u is the stage input, y is the stage displacement and n is the measurement noise.

In the closed-loop, stage displacement $y = Tr - Tn$, tracking error $e = Sr + Tn$ and control effort $u = KSr - KSn$, where $S = 1/(1 + GK)$ and $T = GK/(1 + GK)$ are the sensitivity and complementary sensitivity transfer function, respectively. The sensitivity transfer function S should be small up to the bandwidth frequencies, so that the tracking error e is small. The complementary sensitivity T should remain close to 1 up to tracking bandwidth for good tracking and small in the high frequency region where noise is prominent. Also, the designed controller K should be such that it makes the closed-loop transfer function KS small in the bandwidth frequencies, which in turn bounds the control effort u .

In this approach the design objectives of large tracking bandwidth, good positioning resolution, bounded control effort, sufficient environmental disturbance rejection and adequate measurement noise attenuation are simultaneously incorporated in an optimization problem. The weighted tracking error, $z_s = W_s(r - y)$, weighted control output, $z_u = W_u u$, and weighted displacement $z_t = W_t y$ are the signals that corresponds to tracking bandwidth, control effort bound and noise attenuation objective. The optimal controller K is obtained by solving the optimization problem, $\inf_K \|T_{wz}\|_\infty$. Where, T_{wz} is the matrix transfer function from exogenous inputs $w = [r \ n]$ to the regulated output $z = [z_s \ z_t \ z_u]$ and $\|\cdot\|_\infty$ is the H_∞ -norm.

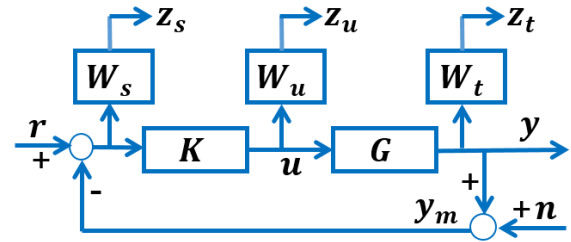


FIGURE 16. The closed-loop transfer function block diagram for mixed-sensitivity minimization is shown. Here, G is the fitted stage model, reference signal r , stage input u , measurement noise n , design weights W_s , W_t , and W_u , and regulated outputs $z = [z_s \ z_t \ z_u]$.

In terms of closed-loop transfer functions the optimization problem can be restated as,

$$\inf_K \|[W_s S \ W_t T \ W_u KS]^T\|_\infty$$

The design weights W_s , W_t , and W_u are chosen so as to give the closed-loop transfer functions S , T , and KS a particular shape to achieve the design objectives. As for example, W_s is chosen to be a low pass filter such that S gets the shape of high pass filter, i.e. S is small in the frequency up to the tracking bandwidth which makes the tracking error e small. The -3 dB bandwidth of a closed-loop system is the minimum frequency at which S becomes -3 dB. So, -3 dB bandwidth of ω_p would mean that the magnitude of the tracking error would be less than 70% as long as the reference frequency is less than ω_p . In case of X-ray optics nanopositioner, -40 dB bandwidth that corresponds to 1% tracking error would be more sensible way of defining closed-loop performance. W_t is chosen to be high pass filter such that complementary sensitivity T gets the

shape of low pass filter, which provides good tracking in low frequencies and good noise attenuation at high frequencies. An adequate attenuation of measurement noise would result better positioning resolution. The design weight W_u effects the shape of the KS transfer function which in turn bounds the control effort. This prevents controller saturation and ensures safe operation of the system hardware.

RESULTS

A H_∞ -controller was designed for the diagonal XX-stage with 15 Hz -40 dB bandwidth.

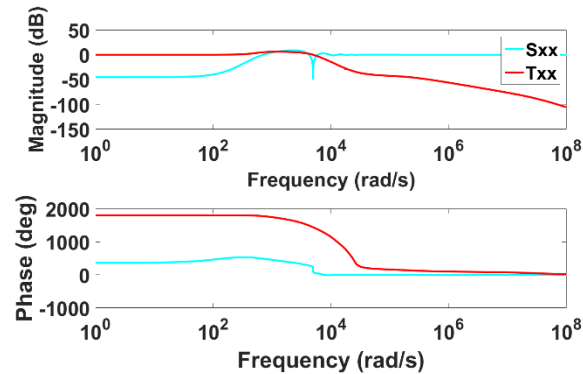


FIGURE 17. The closed-loop transfer function S_{xx} and T_{xx} for the diagonal XX-stage is shown above. The S_{xx} crosses the -40 dB line at 15 Hz and -3 dB line at 95 Hz. The frequency at which T_{xx} crosses the -3 dB line is 763 Hz.

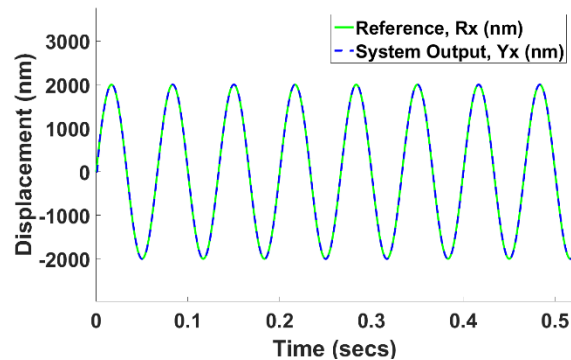


FIGURE 18. The X-stage displacement (blue dashed line) closely tracks the reference (green solid line) sine wave with 15 Hz frequency and 2000 nm amplitude in closed-loop.

The bode plot of the closed-loop transfer functions S_{xx} and T_{xx} are shown in the Figure 17. Figure 18 shows the reference tracking of the XX-Stage in closed-loop. The tracking error is less than 1% since the sinusoidal reference have a frequency of 15 Hz which is within the -40 dB bandwidth of the controller K_{xx} .

SUMMARY

The paper shows the making of an X-Ray microscope that reflects the requirements of the APS upgrade. The identification of the diagonal and the off-diagonal stages of the PI XYZ stages are detailed. The controller for diagonal XX-stage shows good tracking performance. The hysteresis and resolution experiments are not shown here. The parametric model fitting, controller design, and controller implementation for the remaining SISO diagonal stages and MIMO stage will be carried on in the usual manner and presented in subsequent articles.

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ON THE USE OF MICROCONTROLLERS IN MECHATRONICS

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ABSTRACT

Mechatronic systems need feedback control. Controllers for high-end systems become more complex to achieve increasing levels of system performance by compensating non-ideal mechanics and physics. Control approaches and control hardware and software developed for high-end systems trickle down to less high-end systems, reaping benefit of the state-of-the art.

On the other hand, consumer electronic products, adopted by the maker movement, make available lower cost, lower performance hardware and software that is being used for robots, drones, 3D printers and similar mechatronic systems. We are at the point that Arduino microcontrollers can be used to control a mechatronic test set-up, that Inertial Measurement Units (IMU) developed for phones and used for drones can also be used to log acceleration levels in equipment, and that sensors for computer vision are more cost effective than ever before, even including processing hardware.

Based on a number of recent examples, this paper will illustrate the use of microcontrollers and related sensor- and actuator systems for mechatronic system development and -diagnostics. The paper does not have the ambition to give a complete overview, rather it hopes to inspire the mechatronic community in using cost-effective hardware and software to support system development.

INTRODUCTION

Microcontrollers are different from microprocessors in that they carry volatile and non-volatile memory as well as programmable I/O peripherals on-board. Microcontrollers are designed to serve in embedded control applications. Texas Instruments made the first microcontroller called TMS 1000, which launched in 1974. Since then, a lot has happened.

Microcontrollers have been and are extensively used in embedded applications where computational needs are not very high, such as keyboards, remote controls, phones, cars, etc. Whether 4, 8, 16 or 32 bit, MCUs can be bought for much less than \$1.



FIGURE 1 Sample overview of microcontroller- and microcontroller system names

The maker movement has picked up on the versatility of the microcontrollers to control DIY projects. Microchip PIC, Atmel AVR but maybe especially Arduino (see *FIGURE 2*) have accelerated the use and adaptation of microcontrollers and microcontroller boards. They have created communities of enthusiasts around the world, sharing knowledge, work products and inspiring each other for new developments. With Arduino and others, microcontrollers are available on boards that provide all I/O and peripherals, specifically a USB connection for programming and communication with a host, and clock and power circuitry. The boards offer standardized pinning for the I/O, that enables the use of extension boards or so-called *shields*. By stacking shields, peripherals can be added ranging from PWM motor controllers and Ethernet connectors to GPS devices and GSM phones.

The availability of standardized expansion connectors in combination with a wide variety of

shields is a factor contributing to the popularity of the microcontroller boards. Many variations are available, see *FIGURE 1* for an impression (but don't hesitate to use Wikipedia to find more specific information).



FIGURE 2 Arduino Uno R3 (By SparkFun Electronics from Boulder, USA - Arduino Uno - R3, CC BY 2.0)

Now, we see the maker movement reach back into the professional workspace. Makers that are also professional engineers take the benefits of the microcontroller boards and their ecosystems back to their day-time jobs, using them for prototyping and test set-ups. Given the continuous increase in microcontroller capabilities, it is to be expected that professional use will grow beyond prototyping and testing.

The Raspberry Pi has created another revolution whereby computers with microprocessors are scaled down. With standardized I/O expansion and I/O boards, the Raspberry Pi boards are also used for microcontroller functionality.

CASE: MEASURING AND LOGGING OF WAFER ACCELERATIONS

VDL ETG is responsible for integrated wafer handler modules for semiconductor lithography equipment. In addition to thermal conditioning of the wafer substrates, also geometric pre-alignment is performed, bringing positional uncertainty back from ± 1 mm, $\pm 180^\circ$ at the input to ± 25 μ m, ± 150 μ rad before handing over to the exposure wafer stage.

During the transition from the pre-alignment unit to the wafer stage destination, the wafer is held by the end effector, by means of vacuum or

electrostatic forces – depending on the type of lithography equipment. During the transfers, the end effector is mechanically docked to assure maximum repeatability.

Now, during the investigation of a certain issue the question was raised on whether the wafer was slipping on the end effector, which could be caused by too high (in-plane) accelerations. It was concluded that recording of the actual wafer acceleration levels would be a good experiment. Commercial solutions for such measurement do exist, but are expensive – too expensive for our first-order experiment.

We found an open-hardware sensor and logging unit for measuring acceleration levels on humans, to identify biomedical conditions (see *FIGURE 3*). The project is called Totem Open Health [1] and the sensor is a very small form factor combination of

- An MPU-6050 IMU from InvenSense, containing a 3-axis MEMS accelerometer and a 3-axis MEMS gyroscope;
- An nRF51822 multi-protocol single chip for wireless solutions from Nordic, developed for ULP wireless applications with a multi-protocol 2.4GHz radio and a 32-bit ARM Cortex M0 processor;
- A micro-SD connector;
- A battery power supply.

The hardware design is made available to the community under the CERN Open Hardware License. The software and firmware is made available under the GNU GPLv3 license.



FIGURE 3 Open Source 3-axes acceleration and 3-axes gyroscope sensor and microcontroller for logging up to 1 kHz sample rate for up to 40 days (www.wemaketotem.org)

The processor is actually taking care of all I/O: with the IMU, the SD card, and a host that connects over Bluetooth from a smartphone, resulting in a very compact and low power design. Since the design is open, we have redesigned the original PCB to a new version

where the micro-SD connector is on the same side as the other components, facilitating its mounting on a wafer. The sensor allows to measure down to 1 mm/s^2 , see the table.

Table 1 Key data for the MPU-6050 IMU (from [2])

Gyroscope			Accelerometer	
Full scale	Sensitivity	Rate noise	Full scale	Sensitivity
$^{\circ}/\text{s}$	LSB/ $^{\circ}/\text{s}$	dps/ $\sqrt{\text{Hz}}$	g	LSB/g
± 250	131	0.005	± 2	16384
± 500	65.5	0.005	± 4	8192
± 1000	32.8	0.005	± 8	4096
± 2000	16.4	0.005	± 16	2048

We are now collecting our first measurements, trying to prove the hypothesis of too high acceleration levels at the wafer, causing the wafer to slip on the end effector.

CASE: BALL KICKER CONTROL

In the RoboCup soccer Middle Size League (MSL), teams of five autonomous and mobile robots, roughly the size of an office paper bin, play soccer matches (see *FIGURE 4*).



FIGURE 4 Soccer robot utilizing PC based control and microcontroller IMU sensor and kicker (www.robotsports.nl)

The international RoboCup initiative was started in 1997 [3] and encompasses now several

different robot competition leagues. Every year, it organizes a world cup event that involves approximately 500 teams with 3,500 participants from 45 countries. A key objective is to provide a standard problem for research in robotics and artificial intelligence (AI), using collaborative autonomous robots.

Each robot is an extensive mechatronic system with sensing, motion control, high-level control, communication and reasoning.

An important subsystem on the MSL robot is the kicker unit. Most teams use electromagnetically propelled kickers to allow for an adjustable kick speed. To generate sufficient impulse to drive the ball at high speeds up to 16 m/s, high-voltage capacitors (charged typically to 400 V) are connected to the electromagnetic actuators through high-power switches. By modulating the switches, kick control is achieved. In our case, the IGBT switches are controlled by pulses from the microcontroller having a resolution of $1 \mu\text{s}$.

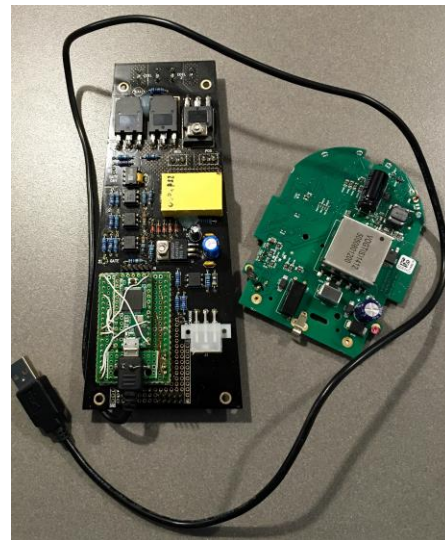


FIGURE 5 Robot soccer kicker system: power board with Teensy for control and I/O, together with 400 V power supply

On the MSL robot of the VDL Robot Sports team, a Teensy 3.1 microcontroller board is plugged onto a custom designed board that holds the power electronics and the necessary optical isolators separating the HV circuit from the rest of the robot, see *FIGURE 5*. The microcontroller provides for a serial (over USB) interface to the control PC and has sufficient computational power and speed to calculate and output pulse sequences with the desired timing

and resolution. The control PC commands the kick speed and elevation, the microcontroller calculates the pulse sequence and controls the power switches. In addition, it handles some elementary safety functionality to eliminate unwanted kick actions.

An interesting side note is that the 400 V voltage source used for charging the kicker capacitor, is actually taken from a personal care device, and that the voltage source itself is controlled by an Atmel AVR microcontroller.

CASE: PROCESSING IMU DATA

For the MSL soccer robot, navigation on the field is key. In addition to an omni-directional color camera, supplementary navigation information is provided by an IMU. While the camera observes field lines, information on the orientation of the robot is required to distinguish between two localization solutions that result from symmetric field lines.



FIGURE 6 Teensy 3.1 microcontroller with MPU-9150 IMU breakout board (in a 3D printed casing).

On the MSL robots from the VDL Robot Sports team, an MPU-9150 integrated IMU from InvenSense (a more recent version of the MPU-6050 with an additional magnetometer in the package) is connected to a Teensy 3.1 microcontroller (see FIGURE 6) which performs the following three tasks:

- Initialization of and communication with the IMU over an I2C bus;
- Performing of sensor fusion calculations for the three sensors in the IMU: gyroscope, accelerometer and magnetometer.

- Accepting commands from the control PC and outputting sensor data in a specific format using a serial connection over USB.

The communication with IMU and host is built using standard available libraries for I2C and serial communication. The Teensy 3.1 [4] is compatible with the Arduino boards and the Arduino development tooling, see FIGURE 7. Libraries that have been developed for Arduino can be used on the Teensy board as well. We have been developing our prototype on the Arduino Due board, with the Arduino development tool, and were able to transfer to the Teensy 3.1 with one “flip of a switch”.



FIGURE 7 Teensy 32 bit microcontroller compatible with Arduino and its ecosystem (www.pjrc.com)

For the sensor fusion, algorithms and libraries are available in the open domain; specifically for the 6- and 9-axes IMUs, algorithms are developed that are optimized for execution speed. In our case, the algorithm developed by Sebastian Madgwick during his PhD research is used [5, 6]. In addition to ground-based robots, the filter is widely used for unmanned aerial vehicles including drones.

Not only is the algorithm for IMU sensor fusion available in the open domain, there are also implementations developed for Arduino and compatible microcontrollers. These implementations can be readily picked up and deployed within hours, with the benefit of community provided documentation and testing.

CASE: FATIGUE TEST SET-UP

As part of our program to research the added value of additively manufactured mechanical parts for precision-mechanical mechanisms, we investigated flexure mechanisms that were manufactured using Selective Laser Melting [7].

An important characteristic of a dynamic flexure is its fatigue behavior. In order to verify our predictions and experimentally determine the influence of various kinds of post-processing of the fatigue behavior, we developed a test setup. The setup excites the flexure near its resonance frequency, allowing for accelerated test sequences. In addition to an off-the-shelf voice coil actuator and actuator controller, we used an Arduino board to control the setup. A microphone was connected to the Arduino board to generate a signal when the flexure broke so the test could be stopped.

Thanks to the Arduino I/O capabilities and its easy-to-learn-and-use development tools, the test setup could be brought to operation within a few days.

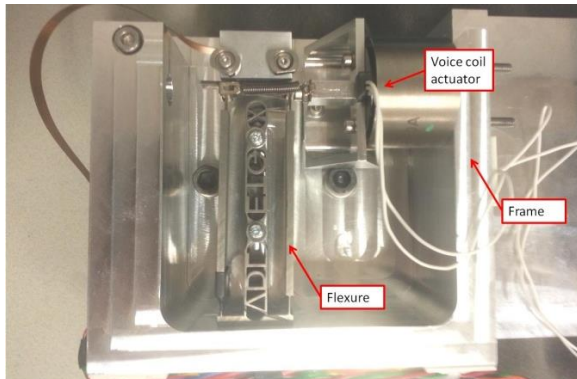


FIGURE 8 test setup to verify fatigue behavior of SLM made dynamic flexures

CASE: DRY ADHESIVE TEST SETUP

Similarly to the flexure test set-up, another set-up has been realized based on an Arduino controller with a motor amplifier and a lead screw motor, see *FIGURE 9*.

The objective of the test is to determine the sticking force of a dry adhesive by introducing a displacement and measuring the displacement and a force.

The Arduino runs a program that controls the lead screw and logs the displacement and the force sensor. Unattended repetitive

measurements are now possible while the measurement data is logged. The development and realization of the test set-up was done in a few weeks, including the controller.

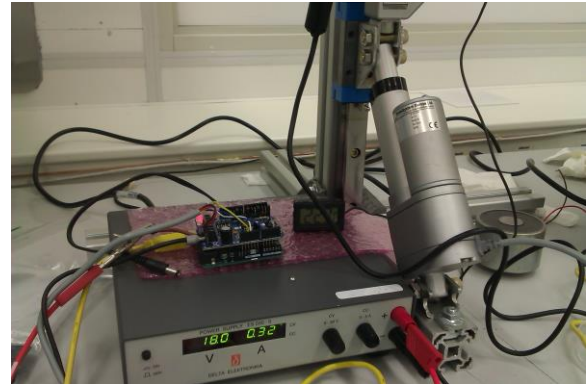


FIGURE 9 Test setup to verify sticking behavior of dry adhesives

DEVELOPMENT ENVIRONMENTS

Most microcontroller applications don't need (and don't run) an operating system. Except for a bootloader that allows the microcontroller to boot and to allow new software to be downloaded over USB from a host, the only code required is your functional code. For an Arduino, the two functions *setup()* and *loop()* must be provided, where *setup()* runs once at boot time and *loop()* runs infinitely after boot.

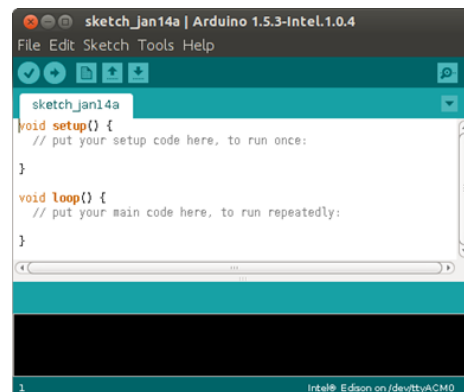


FIGURE 10 Arduino integrated development environment

Given this simple setup, the development tooling for Arduino is relatively straightforward. A graphical environment provides a code window, a serial console to show output from and send commands to the controller, and a status window, see *FIGURE 10*. Importantly, a library manager is integrated that allows to pick up

community provided library packages and use them for development.

The blinking LED example can be considered as the “hello world” for microcontroller developers. Most microcontroller boards have at least one LED (see also FIGURE 6), whereby this LED is connected to a similar port such that the blinking LED example can be readily ported to other microcontrollers.

More versatile development environments and development tooling is available, partially tied in to the specific microcontroller used. An example is MPLAB-X by Microchip, which supports PIC microcontrollers and is expected to also support AVR microcontrollers. Plugins for Eclipse exist, allowing for more professional software development and integration with existing processes.

CONSIDERATIONS

With the Internet of Things as a new driving force, microcontroller development will only speed up more. Devices will become more powerful in terms of computational power, but also gain in I/O capabilities and power efficiency.

The maker community promotes combined hardware and software development. Powered by cost effective but powerful microcontroller boards, rapid development is possible. The educational value is enormous; with relatively simple means, all domains of mechatronic system control can be addressed. Mechatronic control development is democratized. The professional community is picking up, an example of this is the Arduino support package from MATLAB.

Based on our experiences in RoboCup and our day-time jobs, we have become addicted to the power of the microcontroller boards and the communities supporting them. Given the variety of options, our advice would be to choose one family of microcontrollers and boards, and stick to that choice. This will allow you and your environment to achieve the most leverage. Most families have extensive eco-systems, so a need for switching will not easily arise.

CONCLUSIONS

Microcontrollers have been growing rapidly in terms of processing power and connectivity.

While these developments have been mainly driven from consumer electronics applications, they have benefits for mechatronic system development and design.

Integrated devices are available that combine most I/O options that one would ever need, including wireless connectivity over USB or even WiFi or Bluetooth (Low Energy). In addition, the devices are designed for low power usage, supporting mobile and wireless applications.

Development of software for these devices is supported by a world-wide base of examples, libraries and support, enabling rapid deployment of controls for mechatronic systems, and supporting engineering education.

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Micro-Nano TLC Overview of Research in Precision Micro- and Nano-technology

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ABSTRACT

This paper lays out a definition of and boundaries for the field of precision micro-nano (MN) engineering. Micro-nano research cuts across nearly all of the conventional macro-scale areas of precision engineering focus, so a clear definition is critical to understand the field. We provide oversight of the major sub-fields in MN precision engineering work, with particular note of the major driving efforts in each of these areas. This overview is intended to both introduce non-experts to the field as well as help stimulate cross disciplinary information transfer.

INTRODUCTION

The micro/nano field is defined by a specific focus on resolving or exploiting small-scale dependent issues that occur in the standard fields of precision engineering. These scale-dependent effects arise as structures get smaller, different forces begin to dominate, and bulk properties no longer fully define device physics [1]. Such scale-dependent effects are the stepping stones to new devices and new performance regimes for micro/nano engineers. As a rough approximation, unique features of the micro/nano field begin to emerge at the submillimeter scale, particularly in the manufacturing and assembly space [1]. The major physics effects generally occur at micron to sub-micron scales [2]. These effects can be found in all of the conventional precision engineering fields: i) manufacturing & assembly, ii) metrology, iii) measurements & characterizations, iv) design, v) controls/mechatronics.

Engineering on the micro/nano spatial regime requires two capabilities, fabrication and metrology [3]. The first capability defines the ability to create features on the correct scale,

and the second defines the ability to observe what has been created. These two prerequisites have generally been the limiting boundaries of the micro/nano field, and advances in these areas are often the primary/initial focus of research in order to demonstrate a new micro/nano property. This is in contrast with macroscopic precision engineering where the metrology and fabrication techniques are significantly more mature so these elements are often relegated to the background, rather than being the primary focus of development.

The capability to both create and measure on these small scales is a relatively recent innovation of the last half-century, enabled by advances in manufacturing [4] and imaging technologies like SEM, AFM [3,4] as both fabrication and imaging tolerances have been driven down through the μm , and into the nm scale in some cases [3–5].

The interest and the unique value of the micro/nano field lies in identifying and exploiting size-dependent physics. This physics can open new performance bounds for existing functional devices/materials and even provide new functionality not available in natural systems or existing engineered systems. . In addition to the new functions, these scale-dependent physical phenomena also provide important opportunities to rethink conventional designs and standard techniques. The core focus in the micro-nano precision engineering field is to turn these small scale physical phenomena into engineering tools and techniques. Each success in this effort expands the language of micro/nano design; in combination with other prior efforts, this provides the potential for a cascade of advances.

Several examples are noted of unique effects accessible in the micro/nano regime. Quantum effects can be exploited in small structures to produce improved sensors via carbon nanotubes (CNTs) [6] and improved additive manufacturing techniques via 2-photon photolithographic processes [7].

Figure 1 shows how CNTs can be integrated into a MEMS devices in order to create high dynamic range force sensors.

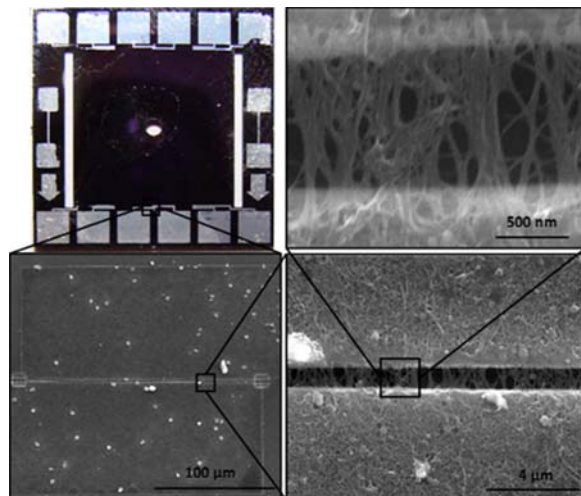


Figure 1. Carbon nanotube-based strain sensors on MEMS device for high resolution force measurements [6].

Two-photon lithography is a direct write technique that relies on nonlinear optical absorption to locally polymerize material within a submicron volume that is smaller than the diffraction-limited spot [7]. A schematic of this process is shown in Figure 2. Nonlinear two photon absorption occurs when two photons of half the energy are near-simultaneously absorbed by the material for the molecular transitions. As the probability of such absorption events is low, high light intensity such as those obtained from a focused femtosecond pulsed laser is necessary for appreciable two-photon absorption. Commonly-used UV curable resists can be selectively cured via near-IR two-photon absorption without any significant single-photon absorption at these wavelengths. Hence, complex 3D parts with submicron features can be fabricated using UV curable resists by translating the high-intensity focal point in space.

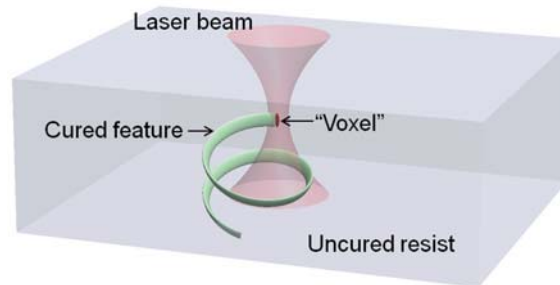


Figure 2. Schematic of two photon polymerization based additive manufacturing [7].

The extreme scaling of structures also drives systems into new operational regimes not generally found in macroscopic system. For instance, Micro-Electromechanical Systems (MEMS) are small scale machines [8–10], whose critical dimensions are often on the μm scale. This small size favors surface over volume effects due to the 2nd order scaling of surface area vs 3rd order scaling of volume [1]. One often-utilized result is that inertia and gravitational loading are relegated to secondary importance relative to electrostatic forces or friction [1]. Such machines have substantially higher bandwidths than their macroscopic counterparts [8,9], and are generally actuated [11–13], and sensed [2,13] by physics or techniques which are unusable at the macroscale.

PRECISION ENGINEERING AT THE MICRO/NANO-SCALE

Micro/nano-scale engineering faces many challenges due to the highly sensitive nature of the environment. Effects which are of secondary importance on the macroscale may be dominant and comparable in magnitude to the signal in the devices operating at the micro/nano scale. This holds true for a variety of physical effects such as vibrations (both acoustic and environmental), thermal driven variations, residual stress-strain, as well as energy and mass flows. In general, on the nanoscale nearly all effects matter [1]. The precision engineering approach of driving down uncertainty through both modeling and proper design is often the best and sometimes only way to carry out micro/nano engineering. This approach includes a focus on determinism in micro/nano engineering through repeatability, measurability, quantifiability and predictability. In practice, the PE focus appears in many forms including i) noise modeling, ii) physics and process modeling to capture variations, and iii)

research on measuring scale-dependent unique micro/nano physics with an eye on repeatability. This committee focuses on the subset of the micro/nano field which utilizes precision engineering principles to advance the field.

AREAS OF FOCUS

Insight into the state of the art in PE for micro/nano can be organized into domains status and latest efforts for each of the major focus areas of PE. This organizational structure aligns with the natural macroscopic analogues to the other PE technical leadership committees. The link across scaling provides a context for the problems and often suggests potential solutions for issues found in the corresponding micro/nano domain. We will cover the unique challenges, solutions and active efforts in each of the domains below.

MICRO/NANO MANUFACTURING

The ability to manufacture a variety of high-quality parts with micro and/or nano scale features within the constraints of rate and cost is essential to fully realize the potential of several promising micro/nano enabled applications [14]. However, achieving this manufacturing capability has turned out to be challenging due to a combination of fundamental and practical limitations that are imposed by fabrication processes and manufacturing equipment on the micro/nano scale. Specifically, these challenges arise due to: (i) lack of well-characterized predictive process models that can capture the essential physics at the micro/nanoscale to aid in process control, (ii) reliance on legacy microscopy instrumentation to implement high-resolution serial fabrication techniques that force one to sacrifice rate and cost of manufacturing, and (iii) the inability to leverage the scalability of traditional semiconductor lithography techniques due to complex 3D geometries and need for multiple non-homogenous materials in the desired parts. Common solution schemes for solving these challenges include: (i) generating validated analytical and empirical models based on process monitoring and in-situ metrology [15], (ii) developing process parallelization techniques that don't sacrifice resolution such as roll-to-roll imprinting [16,17], (iii) and generating hybrid and/or bottom-up manufacturing techniques, such as additive manufacturing, to fabricate multi-material parts with complex 3D structures [18–21]. For example, Saha et al. [7] demonstrate how process models can be used to determine the manufacturing capabilities of a

submicron additive manufacturing technique. Similarly, role of process modeling in nanoparticle position control has been demonstrated by Masui et al. [22] and the role of process control in manufacturing of nanoscale thin films is demonstrated by Singhal et al. [23]. For scalability of manufacturing, parallelization of femtosecond laser fabrication has recently been demonstrated by Wang et al. [24]. To enable fabrication of complex structures, hybrid manufacturing processes have been developed by Chen et al. [25] and Guo et al. [26]. These advances in micro/nano manufacturing rely on the conventional precision engineering principles and at the same time enable practitioners to push the boundaries of precision engineering to reliably fabricate functional parts with ever smaller features.

MICRO/NANO ASSEMBLY

The assembly domain encompasses efforts to manipulate and place existing structures. The conventional analog is classic manufacturing part assembly, where manual or robotic can gripping action is used to place and release parts where desired. While this is not a trivial challenge on the macro-scale, it is further complicated by, among other things i) the difficulty of controlling handling forces which can easily destroy the part, ii) overcoming surface contact forces that often substantially exceed the weight of the object, and iii) anchoring structures together in the absence of conventional attachment elements such as screws or bolts [1,21,27,28]. Precise modeling as well as control of these conflicting forces and motions is a major challenge, and comprises an area of opportunity for precision engineering. Microassembly is one of the more commercially developed domains. Two common techniques to handle the part are equivalent scale grippers [27], or a reference surface for handling, where the micro/nano structure is transferred to the final location without direct handling [28,29]. Techniques will draw from aspects of both of these, as recent work has involved the use of small scale reference surfaces with controllable vacuum suction for the placement of mm-scale micromirrors with μm -scale AM structures on them [21].

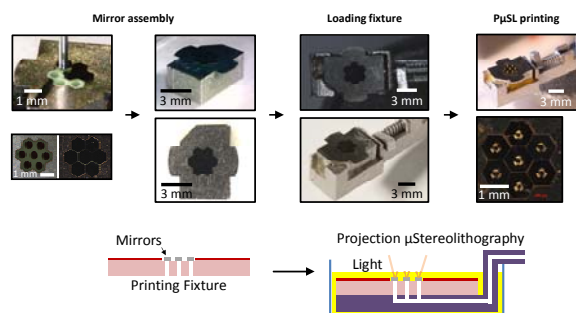


Figure 3. Microassembly of micromirrors with μ AM printed features on them [21].

Some recent microassembly research in the precision engineering field has been focused on all-optical holographic particle handling for assembly of microstructures from discrete particles [30]. This technique draws on light-field control to generate patterns of focused diffraction limited points of light in the working space. These points of light act as stable equilibria locations for sufficiently small particles, as they are drawn into these points via the EM field. Optical handling offers the potential to control the forces down to a scale much closer to that of the materials handled, minimizing the issues of scale-mismatch. Optical techniques also are easily parallelizable and have the potential to run at very high speeds. An example of a micro-assembled structure is shown in Figure 4.

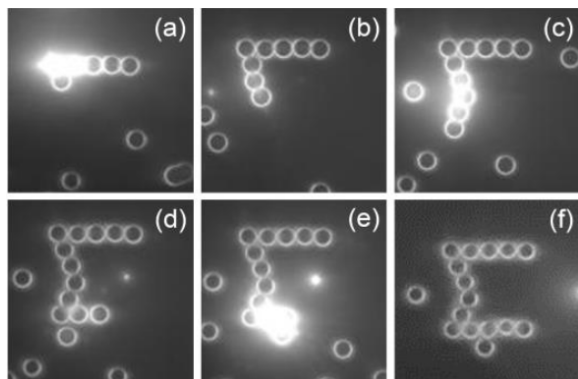


Figure 4. Optically driven microassembled structure [30].

MICRO/NANO METROLOGY

Measurement and characterization of micro and nanoscale features is critical in a number of precision engineering applications including nanopositioning, nanomanufacturing, and precision machining. Currently the biggest challenges in micro and nanometrology are achieving the high-throughputs and resolutions

required for these applications. To improve throughput in micro and nanometrology systems, many precision engineers are developing systems which are capable of making numerous micro and nanoscale measurements in parallel. For example, Cullinan et al. have developed a new tip-based metrology system that uses multiple MEMS chips to make simultaneous AFM measurements over large areas on silicon wafers (See Figure 1) [31][32][33]. This system is capable of achieving wafer throughput rates of better than 60 wafers per hour with sub-nm resolution of features on those wafers [34][35]. Similarly, Chen et al. have developed micro lens arrays which can be used to create multispectral imaging systems that enable rapid microscale pattern detection [25]. To improve the resolution of imaging and positioning systems, precision engineers have developed new hard x-ray based imaging systems capable of achieving sub-20nm feature resolutions [36] and new methods to improve back-focal plane interferometry for nanoparticle position motoring [22]. These types of advances in micro and nanometrology are critical in advancing the field of precision engineering.

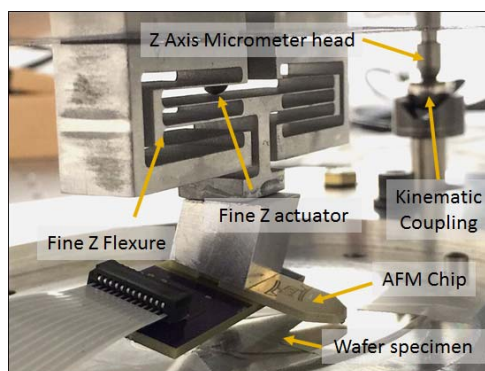


Figure 5. MEMS-based AFM metrology system for measuring critical dimensions on silicon wafers [31].

MICRO/NANO DESIGN

The field of micro-nano precision design deals with the creation of small devices or devices with extremely small features that achieve functional requirements with high repeatability. The design of microarchitected materials that consisted of periodic truss-like lattices made of engineered submicron-sized beam elements that collectively deform to exhibit repeatable bulk material properties is an example of an emerging area within micro-nano precision design. The challenges inherent to this area of design include the issue of managing the high number

of tiny complex 3D features that occur over multiple length scales. Few modeling tools exist that can mathematically cope with predicting the performance of such designs and even fewer tools exist that can optimize their topologies. Additionally, few fabrication approaches exist that can make such materials in reasonable build times, using useful materials, and that occupy large volumes. The tolerances of most fabrication approaches are often unacceptably large to reliably control the bulk behaviors of such materials since the geometries of some of their features are highly sensitive to the overall desired properties. Many of these challenges are currently being addressed using improved modeling approaches and additive fabrication technologies [7,26,37]. A new kind of microarchitected material was recently created that side-steps many of these issues by utilizing micro-actuators, sensors, and control circuitry to exhibit the desired bulk properties using active closed-loop control instead of using the microstructure or its material properties itself to passively exhibit the desired properties [38]. An example 2D lattice of this kind of material that utilizes electrostatic combs as actuators and sensors is provided in Figure 6.

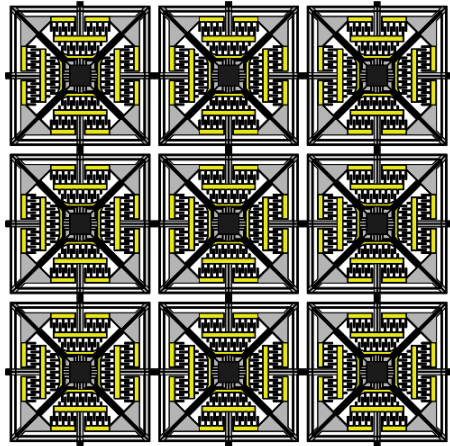


Figure 6. Actively controlled microarchitected material [38].

MICRO/NANO POSITIONING

The field of micro-nano positioners pertains to small positioners that utilize scale-dependent effects to achieve high precision positioning and motion stages. There are a number of challenges that must be overcome by designers of such systems. Fabricating sufficiently small actuators and sensors that achieve the necessary stroke or load capacity is one major challenge. There are only a limited number of

sensors [2,13] and actuators [11–13] that can be successfully operated on these scales. The dynamic range of such systems is also a challenge as well as registration, especially if the positioning system is moved given its limited dynamic range. Calibration is another major issue facing such systems. Recent work has focused on solving these challenges in a number of micro-nano positioning systems, including in-line metrology systems [31]. Additionally, recent work has enabled higher performance micro-mirror positioners [10].

MAPPING EFFORTS

The precision research carried out in ASPE generally covers multiple subfields. For example, micro/nano design is generally closely tied with micro/nano positioners. The chart below seeks to show how some of the presented efforts map to the main active domains, repeated here for clarity: i) manufacturing, ii) assembly, iii) metrology, iv) design, v) positioning.

Table 1. Mapping of active ASPE research to micro/nano subfields.

Paper	i) Mfg	ii) Asm	iii) Met	iv) Des	v) Pos
2pp μ AM [7]	X		X		
In-line metr. [31]			X	X	X
Compound eye [25]	X		X	X	
Holo OT [30]	X	X			
Back-plane Int. [22]			X	X	
Micromirrors [10]				X	X
X-ray microscopy [36]			X		
ECDM [26]	X				
μ Stereolitho [39]	X			X	
μ SLS Stage [40]	X			X	X
Wafer handling [35]			X		X
Comp. plating [37]	X				
Slot-die coating [23]	X		X		

SUMMARY

The micro/nano field is an active and growing field that would greatly benefit from the application of PE principles to create and operate on the smallest scales. The micro/nano field both utilizes and is continually challenged by unique scale dependent effects, which set it apart as an area of research and development. While all sub-fields are showing advances, a recent trend in new manufacturing techniques has driven many advances in micro/nano manufacturing/assembly as well as new material designs to utilize these new techniques.

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Process and equipment driven limits to the performance of two-photon polymerization based submicron additive manufacturing

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SUMMARY

Two-photon polymerization (TPP) is a popular fabrication technique for maskless lithography of functional polymer parts with submicron features. Despite its popularity in fabricating test structures for applications such as photonics, microfluidics, and metamaterials, TPP has had limited adoption as an industrial manufacturing process even for niche low-volume applications. A major bottleneck for this adoption is the lack of process characterization that can be used to either evaluate the performance limits of the technique or to drive the design of improved equipment. As a starting point, here we provide this characterization by identifying the limits of the manufacturing performance metrics imposed by the process and the equipment. Specifically, we characterize the factors that limit achievable rate, surface finish, practical resolution, and scalability. This characterization will be useful to evaluate the performance of existing systems and to design improved systems for two-photon polymerization.

INTRODUCTION

Two-photon polymerization (TPP) is a promising laser-based direct write technique that enables additive manufacturing of millimeter scale parts with submicron building blocks. TPP uses a nonlinear photo-absorption process to polymerize submicron features within the interior of the resists; this provides the ability to control individual volumetric pixel-level features of size 100-200 nm. Due its submicron resolution, there is significant interest in utilizing this technique to fabricate complex 3D structures for applications such as microfluidics, photonics, and mechanical metamaterials [1].

Unfortunately, it is currently not possible to deterministically fabricate functional parts with known certainty on part quality. This is primarily because of a lack of knowledge about the performance limits and fabrication capabilities of TPP. As a result, one is forced to perform a series of trial-and-error based experiments to

identify whether it is possible to fabricate a particular part within the desired tolerances and to select the “right” process parameters. The purpose of this work is to quantify the performance limits of the TPP process so that one may be able to identify the suitability of TPP for fabrication of a specific functional part without relying on ad-hoc experimentation. Specifically, we quantify the practical limitations imposed by the process and equipment on the rate, surface finish, feature size resolution, and scalability of part. This quantification can also be used to drive design decisions to optimize or modify the system and/or process.

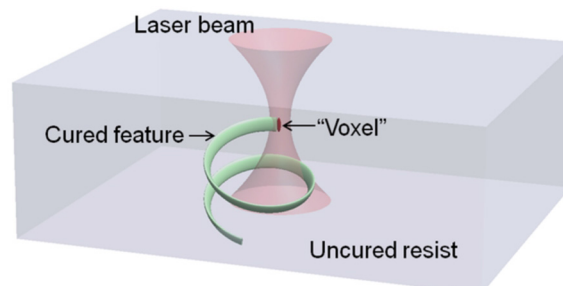


Fig. 1: Schematic of two-photon polymerization based submicron additive manufacturing

PROCESS AND EQUIPMENT

The two-photon polymerization process involves exposing a photo-polymerizing resist to ultra-fast pulsed laser. A schematic of the process is shown in Fig. 1 and the writing parameters are summarized in Table 1. The photo-polymerization process is similar to UV lithography with the primary difference that instead of linear absorption, nonlinear absorption occurs in TPP that involves near-simultaneous absorption of two photons of half the energy [2]. As the two-photon absorptivity of materials is substantially lower than the conventional single-photon absorptivity, high light intensity on the order of TW/cm^2 is required [3]. By selecting photocurable resists that have negligible single-photon absorption at twice the wavelength, it is possible to selectively achieve

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polymerization via two-photon absorption using high-intensity pulsed laser sources.

Due to the nonlinear dependence of two-photon absorption on light intensity, it is possible to localize the polymerization to a small submicron volume around the focal spot. This localized spot (“voxel”) is often smaller than the diffraction-limited laser spot. Thus, features smaller than the diffraction-limited spot can be routinely fabricated using TPP.

Table 1: Two-photon lithography writing parameters

Parameter	Value
Wavelength	780 nm
Pulse duration	~ 100 fs
Pulse repetition rate	80 MHz
Average power	Maximum 60 mW
Writing speed	100 $\mu\text{m/s}$ to 63 mm/s
Stage type	Galvanometric X-Y scanner & piezo Z
Photoresist	IP-DIP (Nanoscribe)

In this study, the commercially available Nanoscribe GT system has been utilized to fabricate parts via a layer-by-layer technique. To generate a single layer, a fast galvanometric X-Y stage is used to scan the laser spot across the resist. The next layer is generated by stepping along the Z direction using a slow piezo stage. As the range of the galvanometric stage is limited to ~140 μm , large millimeter scale parts are generated by stepping in the X-Y plane using a large-motion X-Y stage and stitching together multiple fields generated by the galvanometric stage. Limits to the rate, resolution, and scalability are determined by the physical process and the motion characteristics of these stages.

The stacking of the motion stages in the Nanoscribe system is shown in Fig. 2. The build plate is mounted on an X-Y-Z piezo stage that is itself mounted on a stacked X-Y linear stage. The galvanometric X-Y stage is mounted on a separate frame and positions the laser beam spot in the focal plane of the objective lens. The focal spot is first aligned vertically to the build surface by the microscope objective Z drive and then held stationary during the build process. The build surface is moved vertically with respect to the focal spot by actuating the Z axis of the piezo stage. During stitching of multiple X-Y fields, the build surface is moved in the X-Y plane using the X-Y linear stage. Due to misalignments during mounting, the build surface often lies at an angle to the large X-Y

linear stage. To compensate for the variation in height of the build plane during stitching, the focal spot is vertically aligned to the base of the build plate at the beginning of writing in each field.

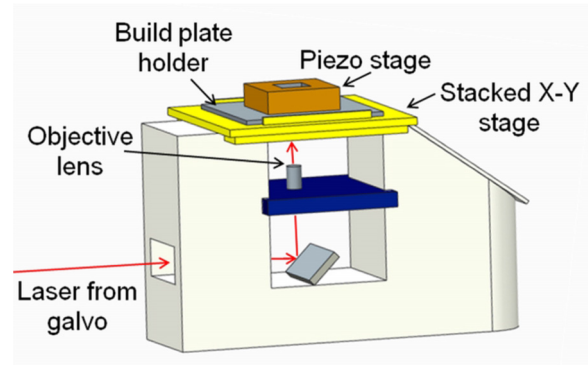


Fig. 2: Schematic representation of the stacking of motion stages in the Nanoscribe GT system.

LIMITS TO RATE

The overall printing rate is determined by two factors: (i) the writing speed at which the laser spot is scanned along a line and (ii) the “dead time” due to acceleration and deceleration of the stages. The relative contribution of these two factors to the overall rate depends on the part geometry and the printing strategy.

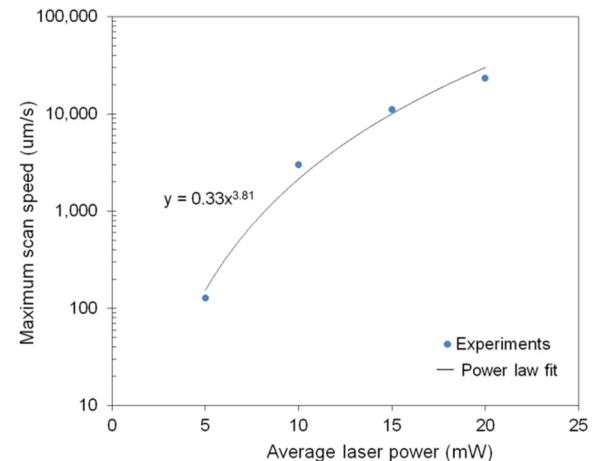


Fig. 3: Maximum linear writing speed versus average laser power. No polymerization is observed above this speed.

The maximum laser scan speed is determined by the polymerization threshold of the resist. In general, no writing is expected at combinations of low power and high writing speed. As the laser power is increased, the maximum writing speed for feasible writing increases. This writing threshold speed is summarized in Fig. 3. It is important to note that the threshold writing

speed is more sensitive to the writing power ($v \sim P^{3.81}$) than the linear power-speed relationship expected from a constant dosage energy estimate. For typical writing power values of > 20 mW, the threshold writing speed is above 23 mm/s. The threshold behavior depends on the nature of the resist and can be tuned by altering its two-photon absorptivity. However, at these speeds the motion systems dominate the overall rate and an increase in the scan speed beyond this value has a relatively small impact on the overall build rate.

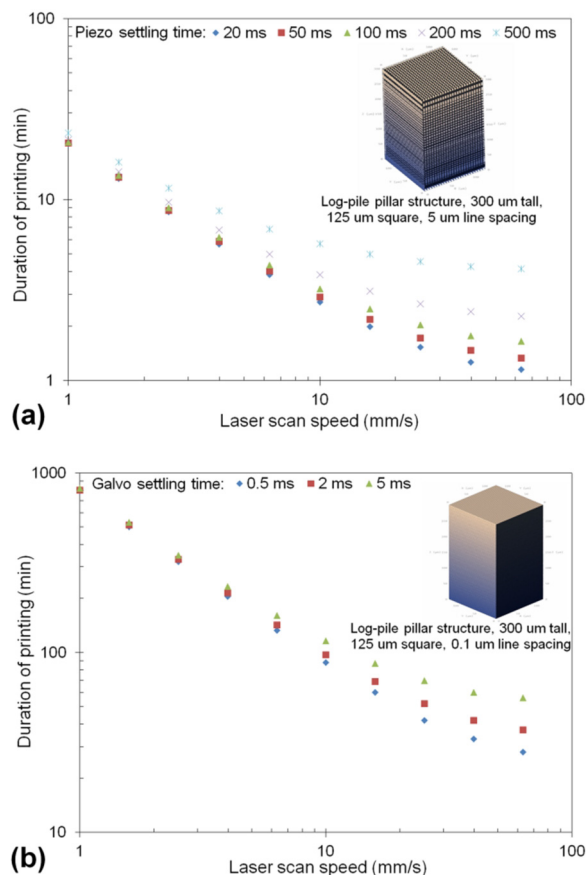


Fig. 4: (a) Effect of scan speed and piezo settling time on duration of printing. (b) Effect of scan speed and galvo settling time on duration of printing

At high writing speeds, the overall printing time is limited by the “dead time” of the galvanometric and piezo motion stages. For parts with sparse features, the layer-to-layer Z stepping time due to the piezo motion dominates the overall rate. Due to the bandwidth limitation of the piezo, each Z-step motion is followed by a piezo “settling time” during which no writing is performed. A similar settling time is used for the galvanometric stage. For dense features, the settling time for the galvanometric stage dominates the overall build rate. The effect of the laser scan speed, piezo settling time, and

galvo settling time on the overall printing duration is summarized in Fig. 4. This printing time was evaluated by the CAM software of the Nanoscribe system and is based on the generated tool path.

LIMITS TO SURFACE FINISH

As micro-scale surfaces in TPP-based AM parts are fabricated by stacking together several individual building blocks, the surface roughness on this scale is determined by the stacking algorithm and the shape and size of the building blocks. During two-photon polymerization, the building blocks (voxels) have an ellipsoidal shape. When a line feature is generated by moving the laser focal spot along a straight line, a linear feature with an ellipse cross-section is generated. The surface roughness is determined by the overlap spacing between successive lines and the width of the line. A schematic of two overlapping line features as part of a surface is shown as the inset in Fig. 5. For our optical system configuration ($NA = 1.4$), the shape of the voxel is such that the major axis is ~ 3.5 times the minor axis. This aspect ratio can be increased by reducing the numerical aperture of the objective lens. For a general ellipsoidal voxel shape, the R_a value of the surface is determined by the line spacing (g), aspect ratio (r), and line width (w) as:

$$R_a = 0.25w \cdot r \left(1 - \left(1 - \left(\frac{g}{w} \right)^2 \right)^{0.5} \right) \quad (\text{Eq. 1})$$

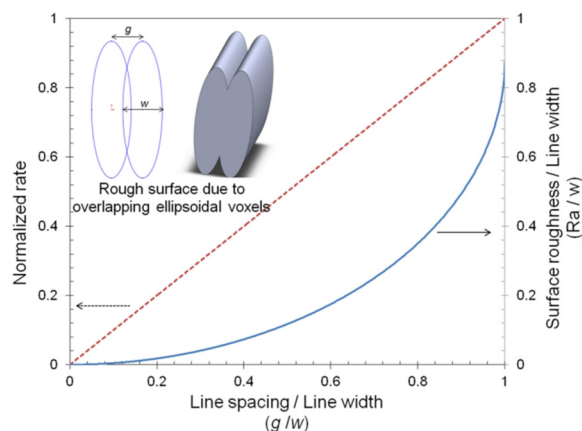


Fig. 5: Surface finish and rate versus line spacing

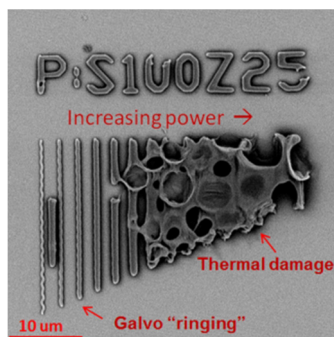
For a line of width 200 nm and aspect ratio $r=3.5$, the surface roughness (R_a) is <1 nm for 90% feature overlap and 99 nm for 10% feature overlap. Thus, when a smooth surface with low R_a value is desired one must closely overlap the line features. However, this strategy reduces the overall fabrication rate as the number of lines to

be printed per unit area increases linearly with increasing overlap (i.e., with a decrease in the line spacing 'g'). This trade-off is summarized in Fig. 5.

It is important to note that this discretization based surface finish model can only provide a theoretical lower limit for the surface finish and underestimates the R_a value at low overlap line spacings. This is because the model is based on the approximation that the individual voxel lines are printed "perfectly" without any deviations in position from their designed values. In addition, it does not account for the surface texture of the polymerized voxels or variations in the width of the voxel features.

LIMITS TO FEATURE SIZE RESOLUTION

When writing along a perfectly straight line, the narrowest lines are obtained by writing at either high scan speed or low laser power, i.e., by writing close to the polymerization threshold of the resist. As the line width is more sensitive to the laser power, often the thinnest lines are obtained at low laser powers (and at combinations of low writing speed). The dependence of feature size on laser power is evident in Fig. 6. In practice, this polymerization threshold line width may not be achievable due to the "smearing effect" of a non-straight laser path. During printing of individual lines, it was observed that wavy lines were generated due to "ringing" in the galvo X-Y stage at low writing speeds. For low spatial frequency ringing, i.e., at high writing speed, this waviness shows up as a wavy line. However, for high spatial frequency, i.e., at low speeds this waviness manifests as a smeared line such that the effective line width is higher than the expected value for perfectly straight laser paths. This smearing effect is evident in the 3rd line from the left shown in Fig. 6. For high resolution writing, one must tune the galvo feedback control parameters to minimize this oscillatory parasitic motion.



LIMITS TO SCALABILITY

The ability to successfully fabricate a millimeter scale part using TPP is determined by (i) the range of the large-motion X-Y stage and (ii) accuracy of the stitching process. For the Nanoscribe system, the range of the X-Y stage is at least 3.5 inches; however, the accuracy of stitching is low and depends on the feature geometry.

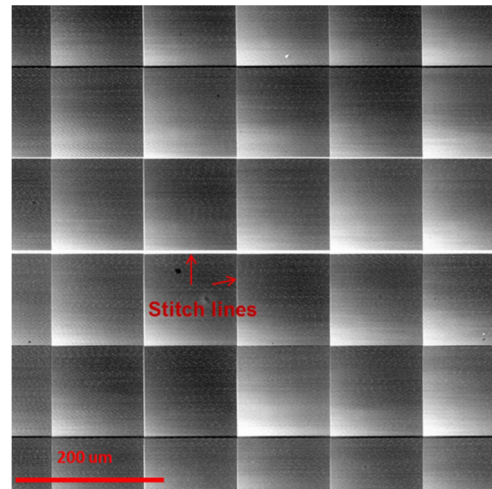


Fig. 7: Scanning electron microscope image of a part fabricated by stitching galvo fields

As stitching relies on slightly overlapping the features in adjacent galvo fields, it is ineffective for parts with sparse features. In such parts, no voxel level overlap is possible due to the inaccuracies in the stage motion. The effect of stage inaccuracies on the effectiveness of stitching is demonstrated in Fig. 7 that shows a single layer of individual voxels printed over a 3 mm X 3 mm area with each galvo field of size 125 μ m X 125 μ m. No field overlap was used for this printing; adjacent fields were printed by displacing the X-Y stage by 125 μ m steps. The stitch lines along adjacent fields can be easily identified by the regions with gaps (brighter regions) or overlaps (darker regions). Each field has a gradient in intensity because of the misalignment between the normal to the build plane and the galvo X-Y scan plane.

To evaluate the effectiveness of overlap stitching, we have characterized the motion accuracy of the X-Y stages. The in-plane parasitic displacement of the two stages was recorded using a capacitance probe as the stages were independently actuated. The results are summarized in Fig. 8. Interestingly, the parasitic errors of the two stages are different; the Y stage has a higher parasitic error (6.1 μ m)

Fig. 6: Smearing of lines written at 100 μ m/s due to a "ringing" galvo stage

than the X stage (1.5 μm). We suspect that this is due to the stacking of the Y stage on top of the X stage. It is expected that the effectiveness of stitching would improve with an increase in the accuracy of the X-Y linear stage.

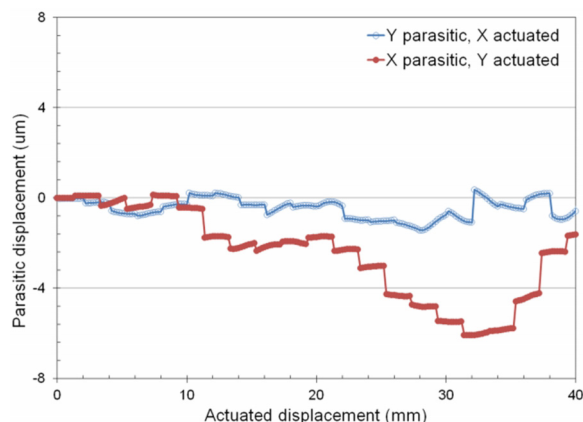


Fig. 8: In-plane parasitic motion of the linear X-Y stage when each axis is individually actuated

CONCLUSIONS

The manufacturing performance of two-photon polymerization is determined by a combination of process and equipment driven limits. Specifically, the following conclusions can be drawn about the performance of TPP:

- (a) The printing rate is determined by a combination of both the resist threshold behavior and the dynamics of the motion stages. Sensitivity of the overall rate to stage dynamics increases with the writing speed. The printing rate also depends on the feature density and varies from 0.01 mm^3/hr to 0.24 mm^3/hr .
- (b) Surface roughness is determined by the shape of the voxels and the voxel overlap strategy. Smoother surfaces can be generated by closely overlapping the voxel features at the expense of the printing rate.
- (c) The practically achievable resolution is limited by the accuracy of the laser scan path as ringing of the galvo stage leads to smearing of the lines.
- (d) The scalability of TPP is limited by the effectiveness of stitching that is in turn determined by the accuracy of the motion stages. In-plane parasitic motion error on the order of 1.5 μm for the X stage and 6.1 μm for the Y stage were observed. One

must overlap fields by at least this amount for an effective stitching.

Based on these observations, one may improve the performance of TPP based additive manufacturing by designing (i) motion stages with higher bandwidth (for rate) and (ii) stages with higher accuracy (for improved feature resolution and stitching). The discretization based writing strategy imposes a fundamental limitation on the surface roughness of additively manufactured surfaces; nevertheless, the roughness can be improved by generating smaller voxel features and by closely overlapping the features.

ACKNOWLEDGEMENTS

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Quick Approach Mechanism for Tip-Based In-line Nanometrology Systems

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Abstract

Rapid setup of tip-based metrology systems is a key factor in enabling these systems to be used in-line with nanomanufacturing processes. This study presents a new automatic approach system that improves the speed and accuracy in which MEMS-based AFM tips can be aligned and brought into contact with the nanometrology features to be measured. The automatic approach consists of a coarse approach and a fine approach step. The coarse motion utilizes the z-micrometer whereas the fine motion control uses voice coil actuator coupled with a flexure bearing in order to give an open-loop motion resolution of 20 nm. With the automatic approach mechanism, the inspection system is capable of cycle times (sample in to sample out) of under a minute, which has great potential for in-line nanometrology inspection.

Introduction

In-line inspection is necessary to screen for faults and errors in nanomanufacturing processes before the wafers move on to the next processing step. However, because in-line inspection is inserted into the production line, such an extra step in process causes extra process cycle time and may harm to the throughput of the system. Therefore, the time required for preparation and imaging is the key point for in-line inspection in nanomanufacturing. In-line metrology is currently done at the microscale using high-speed optical methods but as features sizes become nanoscale, optical methods are no longer feasible. Tip-based metrology systems are able to achieve nm-scale resolutions but the setup time for tip-based methods is too slow for in-line metrology. This paper presents a method for improving setup times in tip-based metrology systems to enable in-line metrology for nanomanufacturing systems.

The in-line metrology system presented in this paper utilizes atomic force microscopes (AFM) that have been incorporated into a single MEMS chip [1]. AFM technique uses a tiny cantilever and make it extremely close to the sample surface. As the distance between two atoms, the atomic interaction become dominant in action. In this stage, the interaction force rapidly increases as those two atoms get closer. Such a characteristic is right the basic of the AFM technique. The single-chip AFMs [1] contain MEMS-based flexures, actuators, and sensors and wire all the components to off-chip circuits for data analysis. This method minimizes the AFM instrument into a single 1 mm x 2 mm chip-sized device.

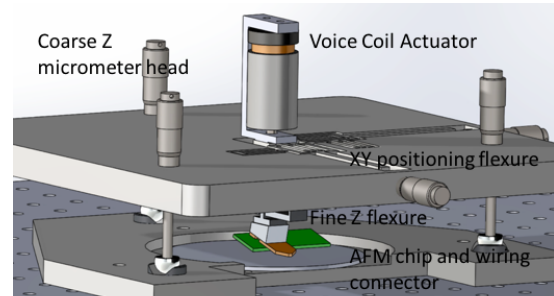


FIGURE 1a. Semantic of Inspection system setup

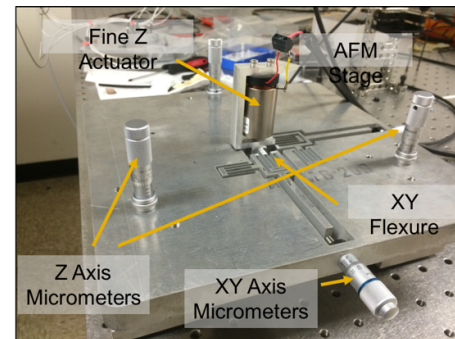


FIGURE 1b. Fabricated in-line dimensional metrology system setup [2]

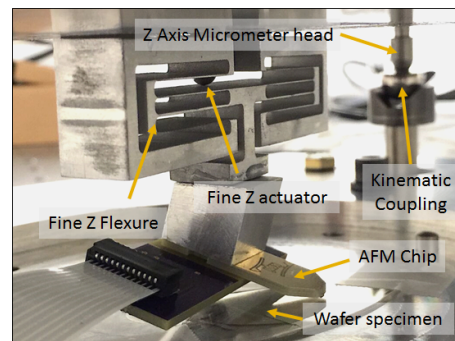


FIGURE 1c. Fabricated Z flexure mechanism [2]

Previous work has demonstrated the ability to use a flexure-based positioning stage and kinematic coupling to position the single-chip AFM metrology system with sub-micron repeatability over mm-scale ranges [3]. However, the operating cycle time is still long for in-line inspection. The inspection procedure can be broken down into four steps: (1) wafer loading, (2) AFM tip approach, (3) AFM scanning, and (4) wafer-unloading. Within those steps, the approach step consumes the most time since a manual operation is required adjust the separation distance between the two stages and position the AFM tip within nanometers of the sample surface.

This paper presents an automated approach procedure where the approach is divided into two steps, a coarse approach step and a fine approach step. The inspection system is shown in Figure 1. It has a 2D AFM-chip positioning stage and a wafer-sample stage, both these two stages constitute the whole system. The coarse approach uses micrometer heads linked with the kinematic coupling to position the AFM tip before it gets the location of approximately 0.5 mm above the sample surface. Afterward, the fine approach uses a voice coil actuator and a flexure mechanism to bring the AFM tip within nanometers off the sample surface where atomic force effects can be "felt" on the AFM tip. Figure 1c shows the fine Z-motion mechanism used in this study.

Fine Z-motion Flexure Design

The system shown in this paper does atomic force microscope metrology using a single-chip AFM produced by ICSPi Corp. Based on the flexure-based positioning stage, the in-line nanometrology system utilizes another flexure mechanism put under the positioning stage for Z-motion. The difference from the flexure in XY direction is the symmetric design in Z-direction. There is no need to be perfectly parallel positioning in XY direction since the deflection in one direction could be compensated by the motion in another one. However, there is no other motion could be used to compensate the deflection made by Z-motion. Moreover, usually the AFM cantilever is much fragile than the measured sample. The cantilever must be as close as the interaction comes into action but also stop translation before hitting sample surface. The micrometer heads have disadvantages such as backlash and manual operation. As a result, this mechanism that adjusts separation between the AFM and sample stages may not be applicable. The system would need a finely motion which has a nano-level step size when AFM unit is approaching to the sample. The micrometer heads are still needed as the kinematic coupling in order to ensure the relative position repeatability, and also can be used for a coarse approach method. As long as the AFM chip is brought to a location near the surface, the approach procedure turns to the fine part.

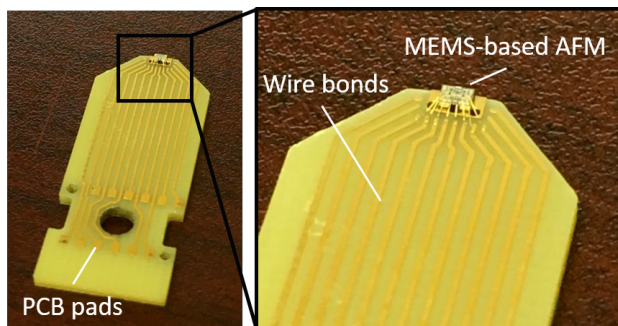


FIGURE 2. MEMS-based AFM chip used in the in-line nanometrology system

The fine approach mechanism enables the system to adjust the height of AFM chip within the range of about 500 μm . In order to constrain the motion on Z-axis, the

system uses two sets of double parallelogram flexures (7075 aluminum) in a symmetrical configuration. The stiffness in tangential and axial directions can be derived by the formula provided by previous study [4].

$$K_t \approx \left[12 - \frac{3}{100} \left(\frac{F_a L^2}{EI} \right)^2 \right] \cdot \frac{EI}{L^3}$$

$$K_a \approx \frac{1}{\left(w^2 + \frac{9}{25} x_t^2 \right)} \cdot \frac{12EI}{L}$$

Because the axial stiffness is much greater than the tangential one, it is fine to treat the parallelogram flexure as a rigid body in this direction. The flexure is attached beneath the XY positioning stage, and coupled with a voice coil actuator (LVCM-025-038-01, MotiCont) which is capable of 11 N continuous force. Figure 3a shows the flexure system produces a displacement of 566.8 μm when subject to the maximum pushing force of the voice coil. Also, static simulation (Figure 3b) shows that the maximum stress generated by this displacement in the flexure structure is about 67.85 MPa resulting in a design safety factor of 7.45. The symmetrical flexure is made by waterjet machining, which utilizes a focused ultrahigh pressure jet mixed with water and abrasives to cut the material. The waterjet may cause some error in beam width because the focused jet cannot consistently provide the same cutting width over through the material thickness. Such an error would make the actual stiffness be somewhat different from the theoretical value. However, later in the actuation procedure, the system introduces a closed-loop control to driving the fine Z-motion, thus the stiffness error may not cause problem in application.

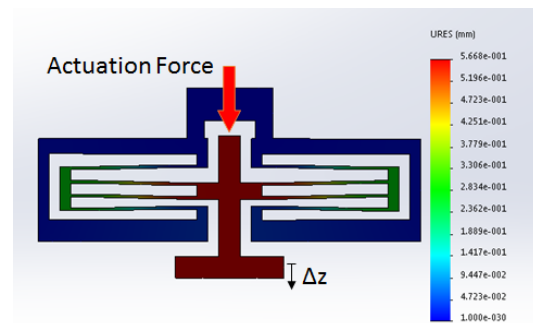


FIGURE 3a. FEA displacement simulation of z-axis flexure mechanism

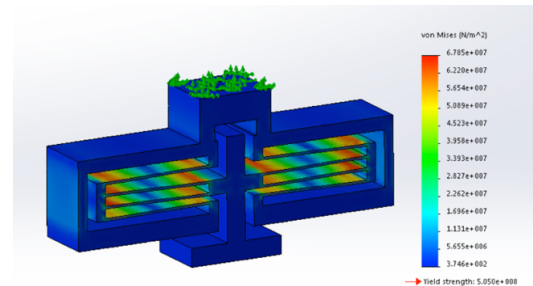


FIGURE 3b. FEA static simulation of z-axis flexure mechanism, the maximum stress when subjecting 11 N

Z-Motion Control

The output force of VCM is generally based on the magnitude and direction of the input current, but usually an output signal of a function generator is controlled by voltage level. Therefore, it needs a voltage controlled current source (VCCS) embedded in the VCM driving circuit, which delivers a current depending on the voltage the user brings to the circuit. In short, the VCCS is playing a role of power amplifier for the VCM, providing a sufficient current for actuating the VCM. Figure 5 shows the VCCS circuit schematic. In the preliminary test, Figure 6 shows that the displacement of fine Z-flexure is linearly related to the applied voltage of voice coil with a sensitivity of $92.3 \mu\text{m/V}$. The electrical noise in the input to the voice coil results in displacement noise of the approach mechanism and thus impacts the resolution of fine Z-motion. The minimum step size in this approach mechanism is 20 nm which is limited by the noise in amplifier circuit. A 20 nm step size is adequate for AFM approach since the cantilever is capable of hundreds-nm movement, some inaccuracy due to the step size may be compensated by the follow-up AFM cantilever.

Figure 7 shows a simple algorithm in block diagram. This study uses three micrometer head (151-255, Mitutoyo), attaching truncated balls on each head end, dealing with the coarse Z-motion. The combination consists of the micrometer heads and vee-blocks ensures the sub-micron repeatability of the AFM location when reloading, also providing rough approach function in range of mm. When the AFM tip is brought to location which is close enough to the sample, usually determined by the AFM signal, the mechanism then turns to a fine Z-motion which drives VCM to generate further sub-mm displacement. Because the coarse approach does not need closed-loop control, but still needs to move far enough to deflect the AFM cantilever at the end of coarse motion, in this study the system is using manual method to coarse approach. In practice of multi-point inspection, each coming sample must have the same height with micron accuracy, such as

FIGURE 4. Closed-loop schematic of the fine Z-motion control in the AFM approach [2]

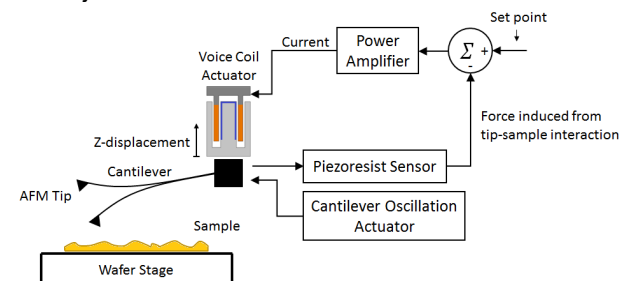


FIGURE 4. Closed-loop schematic of the fine Z-motion control in the AFM approach [2]

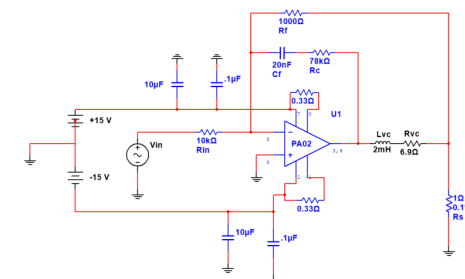


FIGURE 5. Voltage controlled current source (VCCS) circuit schematic, used as a power amplifier in the VCM driving [2]

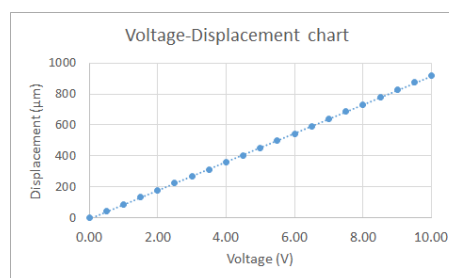


FIGURE 6. Voltage level of the input signal vs. Displacement of the voice coil actuator coupled with the z-axis flexure

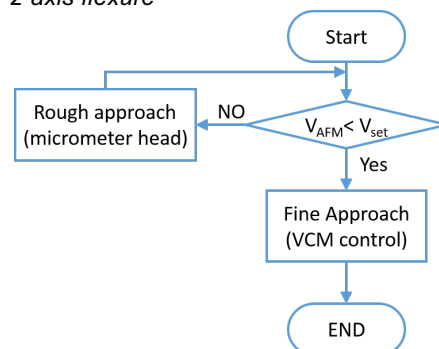


FIGURE 7. Algorithm diagram for the approach

mechanism of the AFM tip

Z-position Stability

The stability of the inspection system is quite important about utilizability. The MEMS-based AFM chip maintains a cantilever oscillation with constant amplitude when doing the measurement. The constant-amplitude mode needs a feedback mechanism embedded in the MEMS design which makes a compensation whenever the oscillation is not running on the set amplitude. In the scanning, the system collects the compensation data on every pixel and exports these data as the topography information. Theoretically the Z-position of the AFM system must be invariant, but in reality any external noise introduced into the mechanical or electrical system will cause a drift to it. If drift amount is making the system too far from the sample, no matter how much it compensates to the oscillation, the cantilever tip will never sense any information from the topography. Figure 8 shows the compensation voltage level varied with time when the system holds on a specific height. At first, the height is adjusted to where that would induce a 1.0 V compensation. The drift issue makes the compensation keep changing and sensing capability becomes expired after 6.5 hours. Such a long period is almost “forever” for the scanning procedure since the MEMS-based AFM chip is capable of fast scanning, which finishes imaging in minute scale.

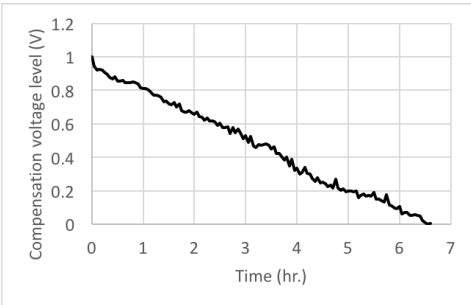


FIGURE 8. Stability test of the in-line nanometrology system. The sensing capability will be invalid after 6.5 hours.

Inspection Time Budget

The time cost in each inspection is the most important specification for in-line nanometrology system. The procedure for AFM operation can be classified into four main steps: (1) wafer loading, (2) AFM tip approach, (3) AFM scanning, and (4) wafer-unloading. Conventionally, the operators spend most of the time for the second and third steps. In order to integrate the AFM into nanomanufacturing system, it is necessary to avoid make a “traffic jam” in front of the inspection system. Thus, the time budget of each step must be mostly reduced.

Previously, the approach only uses micrometer head method for all the approach mechanism. The operator has to always check the AFM signal making sure the cantilever tip is not touching a “wall” and how much distance need to be made further. In this study, the system split the approach to coarse and fine steps. The fine Z-

motion step introduces feedback mechanism and automatic control, which significantly reduces the approach time. With the actuator-driven fine Z-motion, the in-line inspection takes up to 1 minute for an inspection cycle. The AFM approach procedures can be dramatically reduced to about 5 seconds in total, whereas it was 5-10 minutes without fine Z-actuation. Table 1 shows the time budget of each procedures in the inspection. The overall inspection cycle time is less than 1 minute per wafer, which is potentially capable of keeping up with typical nanofabrication, enabling in-line metrology for nanomanufacturing systems.

Further improvement on the budget may be focused on the AFM scanning and wafer exchanging mechanism. Now, this system is using manual method. A wafer handling robot is developing for shortening the wafer loading and exchanging duration. The robot mechanism contains several step motors and an arm equipped with a vacuum suction slot, which can be preprogrammed to execute the wafer transportation job.

TABLE 1. Time budget of each procedure in the inspection

	Wafer Loading	AFM Approach	AFM Scanning	AFM Departing	Wafer Exchanging
Time Budget	7.07 s	5.31 s	26.05 s	0.07 s	20.55 s

Measurement Calibration

Atomic force microscope uses a tiny cantilever to sense the separation between its tip and the sample surface. Once the AFM system gets the information from the first point, it will go to the next one then get the information again. Iteratively gathering the surface information then the stored data will form an image which shows the topography of the inspected area. Briefly speaking, the AFM system makes the cantilever scan onto the sample pixel-by-pixel. Each separation between any two adjacent pixels in XY plane means the lateral resolution of the inspection. And, the intensity of the AFM signal is dealt by an analog-to-digital converter for discretization and restored to data, the pixel separation in Z direction means vertical resolution.

Since the generated image is a bitmap which gathers the intensity information on every pixel, it is necessary to convert the unit of pixel to something with physical meaning such as length in reality. A calibration grating is a good tool to determine the scanning resolution of the AFM. This study uses grating samples (677-AFM and 629-30, Ted Pella Inc.) made by soft (cellulose acetate) and hard (silicon dioxide) samples to do the calibration. Within the two samples, the soft one consists structure of 500 nm line spacing and 100 nm grating height and the hard one is a layer of SiO₂ with 500 nm step height and period of 3 μm. Due to the principle of AFM that is based on atomic integration, the vertical resolution must be somewhat different between hard and soft samples. It shows 0.48 and 1.64 nm/pixel vertical resolution in hard and soft, respectively. Figure 9 shows the scanning result

of both two hardness grating samples in 3D view.

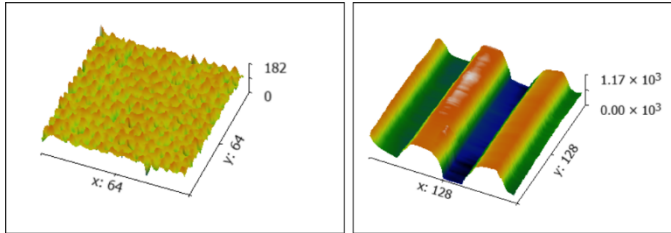


FIGURE 9. AFM test imaging for grating samples which are structured with (Left) grid pattern of 500 nm line spacing and 100 nm height and (Right) 500 nm step height and period of 3 μm

Conclusion and Future Works

This study utilized a fine approach method which divides the approach procedure into coarse and fine sub-parts. Without the fine approach, the approach procedure needs more than 1 minute to get it done. However, the introduction of the fine approach significantly reduces the time to about 5 seconds, which lowers the time budget in inspection and could potentially enable the ability to do in-line mass-inspection in semiconductor nanomanufacturing systems. The stability test shows that even there is drift issue which makes Z-position being varied, there is no worry about invalid of sensing capability. The system needs about 6.5 hours to become out of service but scanning only takes minutes at most for imaging. Moreover, this study also performs a measurement calibration in order to figure out the imaging capability of this in-line inspection system. The vertical resolution is affected by the hardness of the sample. It is 0.48 nm/pixel when scanning on hard surface and 1.64 nm/pixel when soft surface. In order to increase the inspection area of these systems, the next step is to enable the operation of multiple AFM chips on the sample area. As long as the system is able to sense different location at the same time, it should be possible to do investigation of feature distribution among the sample. This will enable quality control to be performed over large areas and be introduced into the manufacturing flow in order to improve yield.

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A HYBRID IMPRINTING PROCESS TO FABRICATE A MULTI-LAYER COMPOUND EYE FOR MULTISPECTRAL IMAGING

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In this paper, we present a hybrid imprinting process that enables the precision fabrication of a multi-layer artificial compound eye (ACE) and its application in multispectral imaging. To achieve high imprinting precision and registration, kinematic couplings (KCs) are installed in the vacuum nanoimprinter [1, 2] to re-position the stamp holder with ~400 nm repeatability. The multispectral imaging system enables effective pattern detection capabilities by precise frequency separation, leading to new applications in space and medical imaging. Imaging experiments on space images and tumor samples were performed to validate the effectiveness of the multispectral imaging system.

MULTI-LAYER COMPOUND EYE DESIGN

Multispectral imaging is a technique to collect different image data at different spectrum, which enables extraction of additional wavelength-sensitive information. Multispectral imaging finds important applications in space imaging [3-5], examination of arts, e.g., paintings [6-7], and biomedical imaging [8-11]. In conventional multispectral imaging systems, different wavelengths are typically separated by filter wheels [12, 13] or a liquid crystal tunable filter [14, 15]. However, such design can only capture one spectral image at a time, making the entire operation time-consuming and the system unnecessarily bulky.

To address the issue, we develop a compact multispectral imaging system based on a multi-channel ACE. As shown in Figure 1, the ACE consists of three structural layers: (1) micro-lens arrays on a convex dome for image formation; (2) a gold light-blocking layer to reject diffused light; and (3) multi-channel color filter arrays, installed

at the bottom of the dome, for frequency separation. A flat photodetector array is placed beneath the ACE for image acquisition. Accordingly, the compact imaging system can simultaneously capture images of different spectral bands.

Since the photodetector array is flat, each micro-lens on the ACE must have different focal lengths to simultaneously focus onto a plane where the photodetector is located. To achieve this goal, a 3 x 4 micro-lens array is designed and divided into four groups, where the diameter of each micro-lens group is 900, 893, 887, 880 microns respectively. The distance between each micro-lens is 2.54 mm.

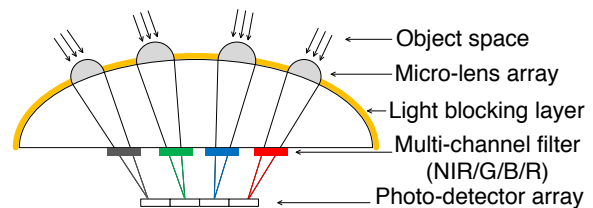


FIGURE 1. Schematic illustration of the multi-channel ACE structures

To investigate the required alignment precision between the individual micro-lenses and the light-blocking layer, ray tracing analyses are performed. Figure 2 presents the analysis results as well as the top view of the ACE with associated micro-lens groups. Ideally, the misalignment of a micro-lens should not cause the circle of confusion to be larger than its original spot size on the focal plane. Based on this criterion, the maximum allowable misalignment is calculated to be less than 1.15 micron.

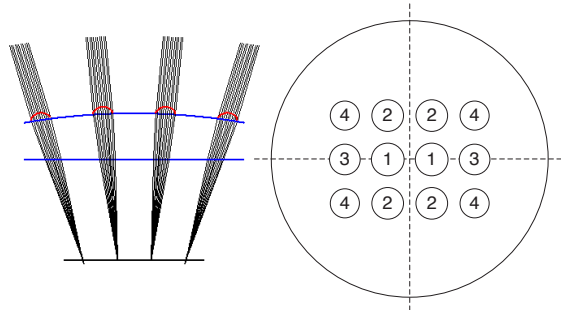


FIGURE 2. Ray tracing analysis and the top view of the ACE with associated micro-lens array groups

HYBRID IMPRINTING PROCESS

Figure 3 presents the hybrid imprinting process. In step 1 - 2, microcontact printing is used to generate the patterns on the light blocking layer via a custom-designed vacuum nanoimprinting system [1, 2]. In this process, a metal ring holds the polydimethylsiloxane (PDMS) stamp carrying distortion corrected micro-lens patterns is installed to separate the printing chamber into two regions; the pressure difference between the top and bottom chambers drives the PDMS stamp to make conformal contact with the gold coated dome (i.e., a plano-convex lens with a diameter of 20 mm and a focal length of 50 mm). After etching, the light blocking layer is completed. In step 3 - 5, the dome is spin-coated with SU8-2010. Following the same procedure, the patterns on the PDMS stamp are transferred to the dome, where each micro-lens is aligned with the light blocking layer with precision ($< 1.15 \mu\text{m}$). After UV light exposure and demolding, the micro-lens array is completed. In step 6, the color filter array is fabricated via photolithography on a thin glass substrate sequentially. The completed filter array is bonded to the bottom of the ACE, aligned with the micro-lenses. It is worth to note that in the imprinting process, the same PDMS stamp is used to pattern both the gold layer and the micro-lens array, which eliminates the alignment need between the different stamps and the metal ring.

To achieve the required registration precision between the micro-lens array and the light-blocking layer, KCs have been designed and installed on the PDMS stamp holder, as shown in Figure 4, where three V-grooves, each spaced 120° apart, are paired with three truncated balls threaded to the stamp holder to achieve exact constraint. The required preload is automatically generated by the atmospheric

pressure when the vacuum chambers begin to pump down, resulting in an estimated preload force of $\sim 870\text{N}$ in normal printing conditions. The detailed design parameters of the KC are summarized in Table 1. Repeatability experiments have been performed, i.e., the stamp is lifted and repositioned for error measurement for 30 times. The repeatability of the stamp holder is calculated to be $\sim 400 \text{ nm}$.

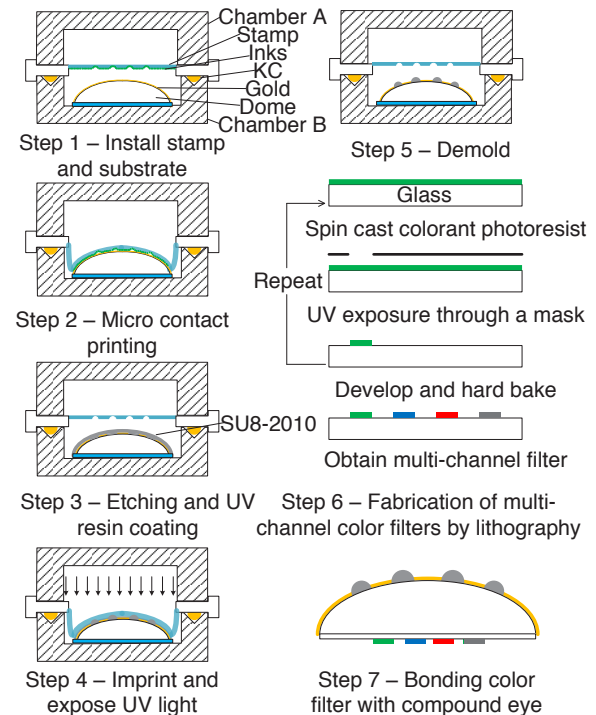


FIGURE 3. Illustration of the hybrid imprinting process for fabricating the multi-layer ACE

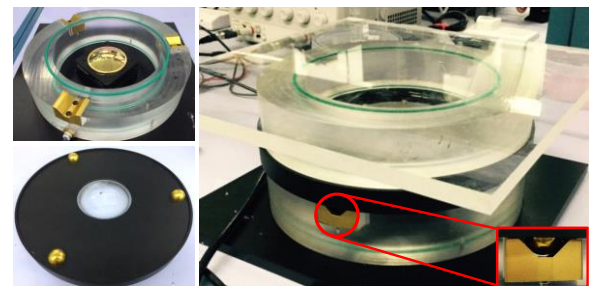


FIGURE 4. Custom-built vacuum nanoimprinter with integrated kinematic coupling

TABLE 1. KC design parameters

Coupling diameter	200 mm
Ball / V-groove materials	304L stainless steel
Ball radius	10 mm
Groove angle	90°
F_{preload}	$\sim 870 \text{ N}$

MULTISPECTRAL IMAGING SYSTEM

Figure 5 presents the optical configuration of the multispectral imaging system where the object plane is located between a broadband light source (400 - 1000 nm) and the ACE. The photodetector is a high-resolution CMOS sensor that captures images formed by the micro-lens array.

Figure 6 presents the measured transmission spectrum of the four-channel filter array. As the color filter array does not block infrared (IR) light, an IR cut-off filter was installed behind the filter array, as shown in Figure 5. Accordingly, frequency separation is achieved via the four different color bands, including 400-500 nm (blue), 500 -600 nm (green), 600-800 nm (red), and 800-1200 nm (NIR).

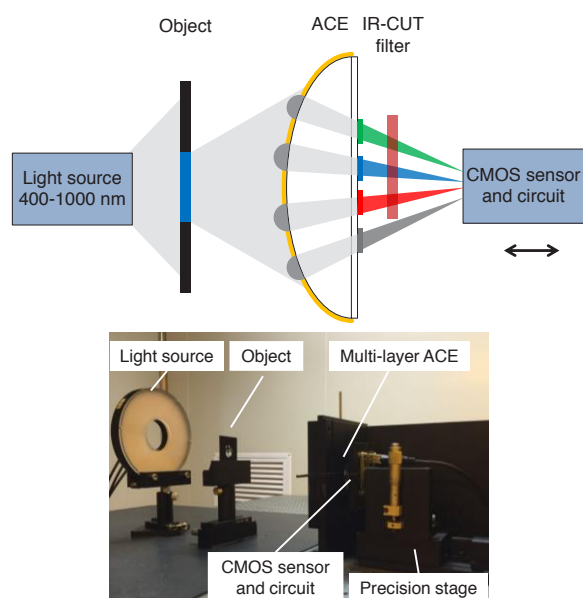


FIGURE 5. Optical configuration of the multispectral imaging system

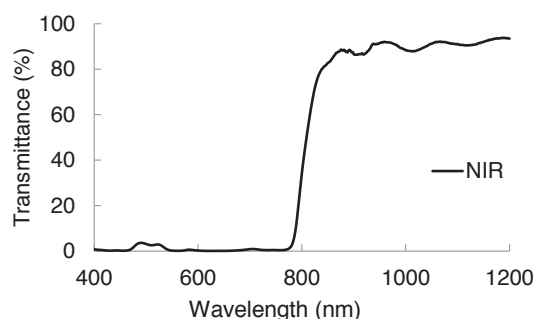
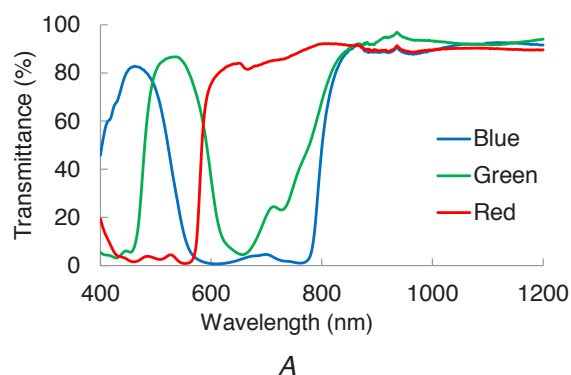


FIGURE 6 Transmission spectrum of red, green blue filters (A); and near-infrared (NIR) filters (B)

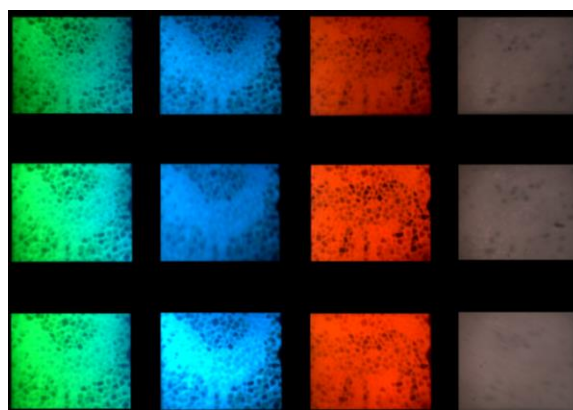
IMAGING EXPERIMENTS

In this section, we perform three multispectral imaging experiments to demonstrate frequency separation capability and practicality of the system, including imaging results on a (1) color blindness test card, (2) space image, and (3) tumor image.

Figure 7 presents the imaging results on a color blindness test card (Figure 7A). Four groups of images in Figure 7B are obtained simultaneously. From the results, we find the “cow” figure can be observed from the red band; and the “chicken” figure can be observed from the green and blue bands, indicating good frequency separation capability.



A



B

FIGURE 7. Multispectral imaging of a color blindness test card (A); and imaging results (B)

In the second experiment, we demonstrate the applications of multispectral imaging in space imaging. As every object has its unique frequency spectrum, object identification can be achieved by combining and processing information from different spectral bands. Figure 8 presents the imaging results on a satellite image of earth (Figure 8A), containing information of vegetation, water and soil. Figure 8B shows the imaging results, where the ocean appears to be black, dark green, and bright blue in the R/G/B channels respectively. Similarly, the land appears to be bright red, bright green and bright blue in the R/G/B channels respectively. These results indicate our compact ACE-based multispectral imaging system may find applications in remote sensing.

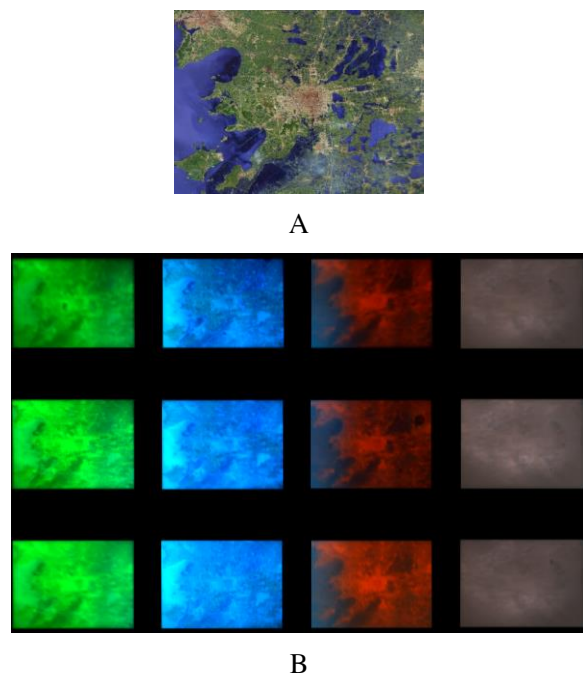


FIGURE 8. Multispectral imaging of a satellite image of earth (A); and imaging results (B)

In the third experiment, we demonstrate the applications of multispectral imaging in tumor screening, showing healthy and cancerous tissues can be distinguished by studying their spectral information. Figure 9A - 9D present the imaging results on four common cancerous tumors (prepared on glass slides stained with haematoxylin and eosin), including lung adenocarcinoma (A), gastric carcinoma (B), colorectal carcinoma (C) and breast carcinoma (D). The yellow lines in the optical images circle the tumor site; the black dashed lines indicate

the field of view of the multispectral imaging system. From the imaging result, it can be observed that different tumors appear to have stronger contrasts in specific color bands. For example, lung adenocarcinoma is sensitive to the blue band; and gastric carcinoma is sensitive to both the red and green bands. These results confirm the practicality of multispectral imaging in tumor screening.

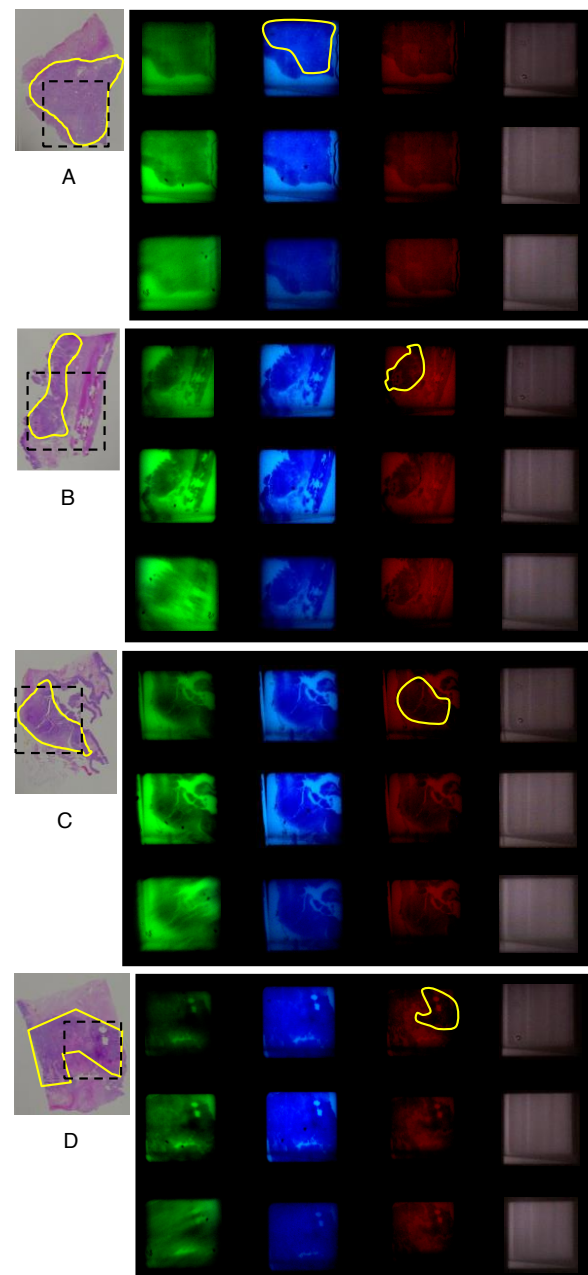


FIGURE 9. Multispectral imaging of cancerous tumors, including lung adenocarcinoma (A), gastric carcinoma (B), colorectal carcinoma (C) and breast carcinoma (D)

CONCLUSION

We have developed a hybrid imprinting process to fabricate a multi-layer ACE with non-homogeneous micro-lenses using a new vacuum nanoimprinting system, where KCs have been designed and installed to achieve ~400 nm repeatability and 1.15 micron registration accuracy on a nonplanar substrate. Based on the multi-layer ACE, a compact multispectral imaging is developed to perform imaging experiments on a (1) color blindness test card, (2) satellite image of earth, and (3) various cancerous tumor samples. The results confirm the frequency separation capability and practicality of the system. Better results may be achieved by increasing the number of color bands and reducing the bandwidth.

ACKNOWLEDGMENT

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Additive Fabrication and Micro-Assembly with Scanning Holography

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ABSTRACT

The aim of this research is to demonstrate a new optical system that relies on scanned holograms to both (1) improve the parallelization and throughput of conventional laser-scanned lithography systems and (2) expand the working range of traditional holographic optical trapping systems. Scanning lithography has been used to fabricate periodic structures without masks, but we introduce a means to combine the benefits of fast-scanning galvanometers and phase-only holography for rapid and parallelized direct-writing in a photopolymer resin. Our approach also allows for time-shared holograms to create complex light fields, including multiple optical traps and optical vortices, over a large range for the purposes of trapping and manipulating microscale particles. These advances further improve the flexibility and parallelizability of optical additive fabrication techniques.

INTRODUCTION

Optical fabrication techniques including scanning lithography [1-3], scanning optical trapping (SOT) [4-5], and holographic optical trapping (HOT) [6-7] provide a means to additively fabricate and manipulate objects on the micro- and nano-scale. Each of these methods typically relies on either a fast-scanning mirror galvanometer or phase-only spatial light modulator (SLM) to achieve beam steering. Our approach couples both a phase-only SLM and a fast-scanning mirror galvanometer (galvo) to enable casting of high-resolution, dynamically-reconfigurable optical fields over larger areas than conventional scanning-only or holographic-only techniques.

The focus of this work is to introduce a fast and parallelized method for both (1) direct writing of periodic microlattices and (2) to expand the working area of conventional holographic optical tweezers (HOT) via time-shared holograms. The following two sections will discuss contributions to lithography and optical trapping separately.

The flexibility of the proposed scanning holographic lithography (SHL) and scanning holographic optical trapping (SHOT) approaches will enable a variety of applications. Holographic direct-writing in a photoactive resin will allow for rapid additive fabrication of periodic microlattices, which could be used as microarchitected metamaterials with tailored mechanical [8] or photonic [9] properties. Additionally, scanning holographic stands to improve the working area of optical trapping systems, which would enable optical tweezers to assemble larger structures consisting of multiple materials.

SCANNING HOLOGRAPHIC LITHOGRAPHY

In a traditional laser-scanning lithography system, a single focused laser spot must cover every point that is to be photopolymerized [10]. However, traditional approach can be time-consuming and inefficient. It has been previously demonstrated that computer-generated holographic (CGH) beam shaping via SLM allows for some degree of parallelization [11, 12], but CGH-based approaches on their own have a limited scan range due to the finite bandwidth of the SLM.

Our SHL approach builds upon conventional single-beam scanning lithography systems by dynamically splitting the input beam via SLM prior to galvo-scanning, allowing for repeating features to be written simultaneously. SHL relies on the large scan area of the galvo mirrors to provide wide-field and parallelized writing.

This research uses an acrylate-based free radical photopolymerization reaction for curing. The photochemistry reported here has been tuned for visible light initiation as 532 nm lasers are available at the long coherence lengths required for high-quality holography. The photoactive resin consists of a trifunctional acrylate (Sartomer SR9035) at a mass fraction of 0.99, safranin dye initiator (Sigma-Aldrich) at a mass fraction of 0.005, and triethanolamine

co-initiator (Sigma-Aldrich) at a mass fraction of 0.005. Solutions are prepared and tested in a darkened room to prevent premature crosslinking induced by overhead fluorescent lights.

The SHL system shown in Figure 1 consists of a single continuous-wave (CW) laser (3W at 532 nm, Laser Quantum Opus), 256-by-256-pixel phase-only spatial light modulator (SLM, Boulder Nonlinear Systems HSP0532-256), and 2D galvo mirror system (Thorlabs GVS012). The galvo mirrors are controlled with two $\pm 10\text{V}$ analog outputs from a standard USB DAQ card (NI USB-6002). The final focusing lens is a doublet lens with $f=100\text{mm}$ focal length (Thorlabs AC254-100-A-ML).

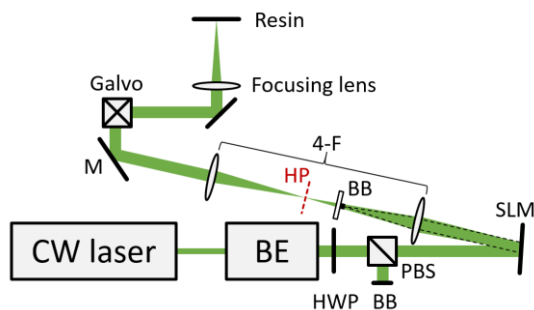


FIGURE 1. Block diagram of the experimental setup. BE: beam expander, HWP: half-wave plate, PBS: polarizing beamsplitter, BB: beam block, SLM: spatial light modulator, HP: hologram plane, 4-F: 4-f telescope with two $f=250\text{mm}$ doublet lenses, M: mirror.

Additionally, a Fresnel beam block is placed at the Fourier plane of the SLM to remove its zero-order undiffracted spot [13]. The desired diffracted orders are shifted downstream of this beam block using a lens phase added to the SLM.

In one case study, we programmed the SLM to split the input beam into five separate focal points in the Fourier plane and we scanned the 2D galvo in a circular motion, resulting in an Olympic flag-style pattern as shown in Figure 2.

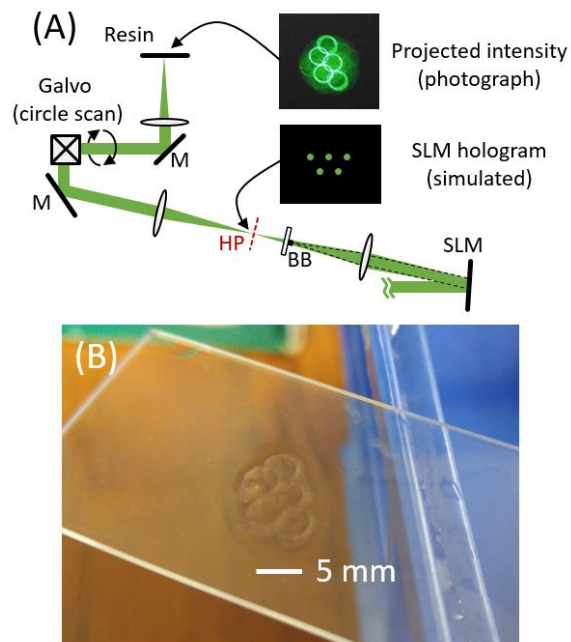


FIGURE 2. Block diagram of the experimental setup demonstrating how five point traps created by the SLM can be scanned to form a series of rings (A). The resulting intensity pattern is cast in a photoresin (B).

The resulting structure in Figure 2B is approximately 10mm by 6mm, but further reduction in size is possible using a final scan lens with more focusing power, such as a microscope objective.

SCANNING HOLOGRAPHIC OPTICAL TRAPPING

Since its debut in the 1970s [14], optical trapping using radiation pressure from high-focused laser beams has proven itself as a powerful and flexible approach for precise application of forces on micro- and nano-scale objects [15].

There are currently a number of drawbacks to the two most common methods for parallelizing optical trapping systems. The first, SOT, relies on a galvo mirror [16] or acousto-optic modulator (AOM) [17] to trap particles over a large range using a single time-shared point and it is limited to one type of trap (e.g. Gaussian point trap, Bessel beam, or optical vortex). SOT systems must scan faster than the time constant for particles to thermally diffuse from their trapped locations. In contrast, HOT systems rely on spatial light modulators to create dynamically-reconfigurable optical fields and are limited by a

proportional relationship between projected working area and optical field feature size.

We propose a SHOT system that combines the large working area of scanning mirrors with the beam shaping abilities of a phase-only SLM. This approach combines the benefits of both SOT and HOT systems, namely the working range of scanning systems and the complex beam shaping capabilities of CGH. The SLM's projected area is much smaller than the working area of the galvo in order to preserve the small features that are created by the SLM. Adjacent traps of any type can be grouped into the same SLM hologram pattern, while the scanning galvo mirrors time-share the beam between holographic groups of traps.

The SHOT system shown in Figure 3 consists of the same laser, SLM, galvo, DAQ, and Fresnel block as featured in the SHL system featured in Figure 1. This system also relies on a 100x oil-immersion microscope objective (Olympus Plan Apo Lambda, NA=1.45) and three-axis micropositioning stage (Thorlabs MAX341) for sample positioning. Microparticles are suspended in an aqueous solution within a sample chamber consisting of a standard microscope slide and cover slip separated by approximately 40 microns.

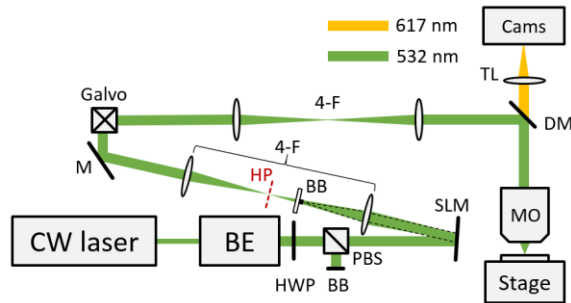


FIGURE 3. Block diagram of the experimental setup. BE: beam expander, HWP: half-wave plate, PBS: polarizing beamsplitter, SLM: phase-only spatial light modulator, BB: beam block, SLM: spatial light modulator, HP: hologram plane, 4-F: 4-f telescope with two $f=250\text{mm}$ doublet lenses, M: mirror, DM: dichroic mirror, TL: tube lens, MO: microscope objective.

The SHOT system creates a wide-field phase pattern by (1) updating the galvo's position by writing new voltages via the DAQ card, (2) updating the SLM phase pattern, and (3) briefly pausing to enforce the positions of trapped particles. An example of a series of phase

patterns, galvo voltages, and the resulting trap groups are shown in Figure 4. The system can complete a full loop through four galvo and SLM updates at a rate of 125 Hz.

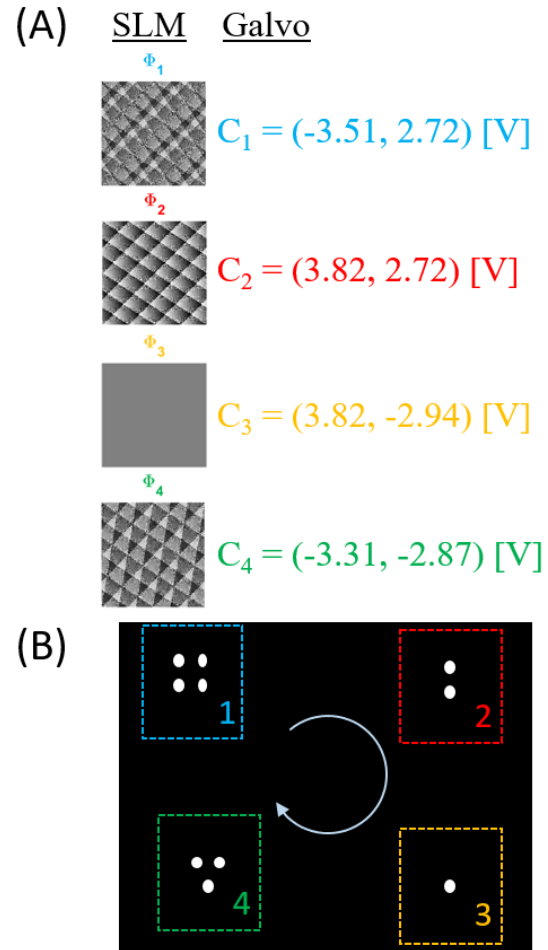


FIGURE 4. Example SLM phase patterns and galvo voltages (A) used to create a series of four trap groups across a wide field (simulated) (B).

We will now briefly describe the analytical tools necessary to rapidly calculate corresponding galvo positions and SLM phase patterns that create a time-shared series of multiple holograms over a large working area. The algorithms in this section allow for nearby traps to be grouped into the same hologram pattern on the SLM. It is important to first identify the maximum range that the scanning galvo can cover, also known as the galvo's working area, which is represented by the bounding black rectangles of Figure 5. Within the bounds of the galvo's working area is a smaller region that can be covered by a single holographic interference pattern, also known as the SLM's working area

(shown as the gray dashed rectangle in the first three steps of Figure 5). Traps within the highest density regions are assigned first and the algorithm iterates for unassigned traps. Once all traps have been assigned to a group, the centroid of each group is calculated and converted to a galvo position voltage. The groups are then reordered into a clockwise orientation about the center to reduce galvo travel time. Finally, the coordinates of each trap about its group's centroid are used to generate the SLM phase pattern for the group. The assignment algorithm only needs to execute each time the overall pattern changes, such as when the user adds, deletes, or modifies the position of a trapped particle.

From the trap group assignments, phase-only, 8-bit holograms are calculated for each holographic group using the adaptive-additive Gerchberg-Saxton algorithm [18] at rates between 150 and 200 Hz. The system scans between trap groups by updating both the SLM and galvo mirrors at rates between 400 and 500 Hz.

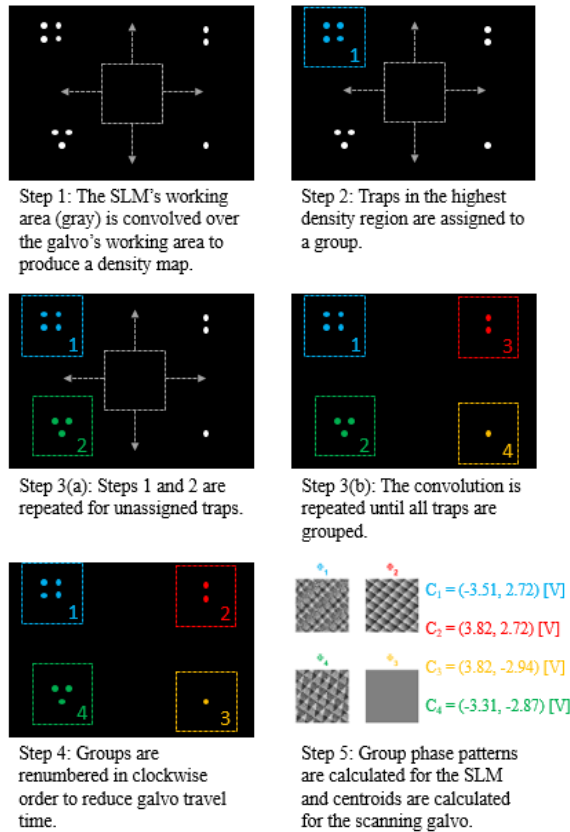


FIGURE 5. Five-step process for grouping adjacent traps into hologram groups. The user

inputs a desired set of traps across the full working area of the galvo and the algorithm outputs time-shared group centroids and phase patterns.

An example of a three-group time-shared array is shown in Figure 6. Future research will pursue automated full-field repositioning algorithms [19].

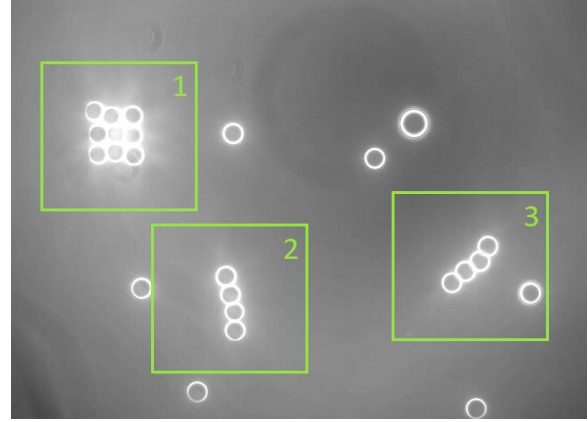


FIGURE 6. Three independent trap groups holding 17 microparticles. The green rectangles represent the projected working area of the SLM, which is time-shared between the three groups: (1) a 3-by-3 array and (2, 3) 4-by-1 linear arrays. All microspheres are 4.21 microns in diameter.

CONCLUSIONS

We introduced a new scanning-holographic optical system that is capable of both (1) parallelized maskless lithography and (2) wide-field optical trapping with high resolution. In a proof-of-concept experiment for the SLH system, an example structure consisting of five adjacent rings was written in photopolymer. To demonstrate the SHOT approach, 17 microparticles were held in three different holographic patterns.

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Theoretical Analysis of Improved Back-Focal-Plane Interferometry for Monitoring Nanoparticle Position

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INTRODUCTION

Recently in order to fabricate 3D polymer nanostructures with high reliability, nano-defects removal processing is attracting great interest. A nano-removal processing method, which has both nano-level resolution and 3D mobility, is needed. Established nano-removal processing are divided into two major kinds, scanning probe microscopy and laser processing. However it is difficult to achieve both of nano-level resolution and 3D mobility.

We have been developing a novel nano-removal processing method by combining photocatalyst nanoparticle and laser trapping [1]. Figure 1 shows a schematic diagram and process flow of the method. Whereas a measurement technique of nanoparticle position is vital to achieve nano-scale processing resolution. Then we focused back-focal-plane (BFP) interferometry, which is one of the most powerful techniques for monitoring nanoparticle position [2], because it requires simple apparatus and provide precise nano-meter, pico-newton scale measurement [3]. However improvement of BFP interferometry is needed to nanoparticle detection with higher resolution.

In this report, we proposed an improvement method of BFP interferometry and theoretically analyzed measurement characteristics of the proposed method.

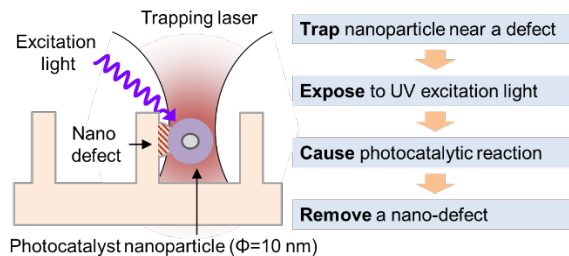


FIGURE 1. Concept of nano-removal processing

CONCEPT OF IMPROVED BACK-FOCAL-PLANE INTERFEROMETRY

Conventional back-focal-plane interferometry

Figure 2 shows a mechanism of conventional BFP interferometry. First a particle is exposed to focused laser beam, and then a part of light is scattered. The scattered light and the unscattered laser beam cause interference in back-focal-plane of collimating lens. Finally intensity distribution of interference, which is detected by a quadrant photodiode (QPD), changes as to a particle displacement.

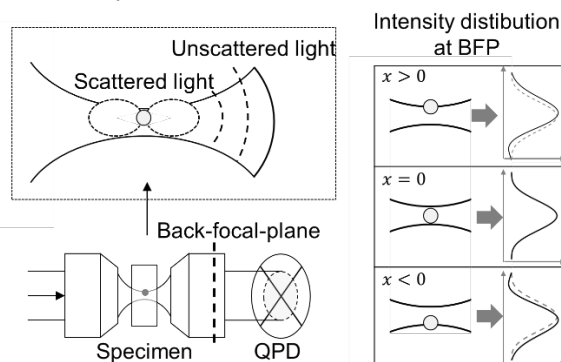


FIGURE 2. Schematic diagram of BFP interferometry

Under some approximations a QPD signal can be analyzed [4]. The approximations are follows: (1) Intensity modulation of a laser beam due to a nanoparticle is negligible. (2) Only first-order interference is considered. (3) A focused beam is considered as a zero-order Gaussian beam. Defining $r = 0$ at the focus, diverging light is observed at angle θ and φ from optical axis (Fig.3). The unscattered electric field at r is

$$E(r) \approx \frac{-ikw_0 I_{tot}^{\frac{1}{2}}}{r(\pi\epsilon_s c_s)^{\frac{1}{2}}} \exp(ikr - k^2 w_0^2 \theta^2 / 4). \quad (1)$$

Here I_{tot} is total power in the beam; k is wave number and ϵ_s , n_s and c_s are the permittivity, the refractive index and the velocity of light in solvent;

w_0 is $1/e$ radius of the focus. The factor $-i$ in Eq.(1) expresses the Gouy phase shift [5] and it provides an important π rad phase jump between the scattered and unscattered light in the far field.

A particle is located in the focal plane and lateral displacement is simplified to a one-dimensional displacement x . The unscattered electric field at this point is represented as

$$E(x) = \frac{2I_{tot}^{\frac{1}{2}}}{w_0(\pi\epsilon_s c_s)^{\frac{1}{2}}} \exp(-x^2/w_0^2). \quad (2)$$

The polarizability of Rayleigh particle is

$$\alpha = \frac{d^3}{8} \cdot \frac{n_r^2 - 1}{n_r^2 + 2}. \quad (3)$$

Here relative refractive index of the particle is $n_r = n/n_s$. The Rayleigh approximation for the scattered electric field at $\mathbf{r}$ is

$$E'(\mathbf{r}) \approx \frac{k^2 \alpha}{r} E(x) \exp[ik(r - \sin \theta \cos \varphi)], \quad (4)$$

at observation angle θ and φ ($\varphi = 0$ along the x axis). The whole intensity in time-averaged light intensity $I(\mathbf{r})$ in the BFP is

$$I(x) = \frac{\epsilon_s c_s}{2} (|E + E'|^2) \approx \frac{\epsilon_s c_s}{2} \{|E|^2 + 2\text{Re}(EE')\}. \quad (5)$$

Here second factor is a relative shift in the intensity caused by the interference and described as

$$\frac{\delta I(x)}{I_{tot}} = \frac{\epsilon_s c_s}{I_{tot}} \text{Re}(EE') = \frac{2k^3 \alpha}{\pi r^2} \exp\left(-\frac{x^2}{w_0^2}\right) \times \sin(kx \sin \theta \cos \phi) \exp(-k^2 w_0^2 \theta^2 / 4). \quad (6)$$

In our approximations any higher order interference $|E'|^2$ is ignored. The whole light, which is acquired by the (+) and (-) halves of QPD along the x axis, can be calculated as integrating $I(\mathbf{r})$ over angles $0 < \theta < \theta_{max}$ and $-\pi/2 < \varphi < \pi/2$:

$$S_{\pm} = \int_{-\pi/2}^{\pi/2} d\varphi \int_0^{\theta_{max}} I(\mathbf{r}) r^2 \sin \theta d\theta. \quad (7)$$

Finally a QPD signal is calculated as $(S_+ - S_-)/(S_+ + S_-)$ using Eq.(7).

For $x \ll w_0$ the response is proportional to d^3/w_0^3 , meaning a sensitivity dependence on a size of a particle and beam waist.

According to the factor $\sin(kx \sin \theta \cos \phi)$ in Eq.(6) intensity distribution of $\delta I(x)$ is symmetric to a direction perpendicular to the x axis. Therefore the intensity shift on (+) and (-) halves is equal and opposite. By contrast unscattered beam has a point symmetry Gaussian distribution. Hence on the denominator $(S_+ + S_-)$ of the

signal the intensity change is canceled and the intensity of the unscattered Gaussian distribution remains. Whereas on the numerator $(S_+ - S_-)$ the unscattered beam counteracts each other and the intensity change remains. As a result a QPD signal can be considered as a relative intensity ratio of unscattered beam and interference.

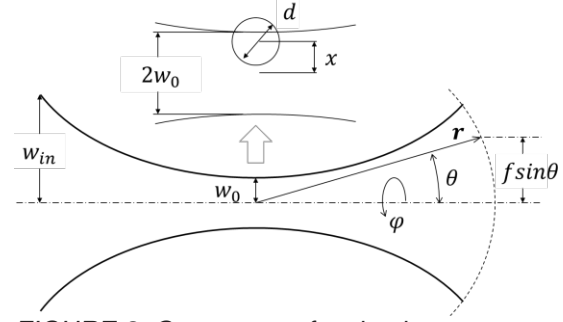


FIGURE 3. Geometry at focal point

Improvement of position detection

We focused on intensity distribution in BFP and proposed improved BFP interferometry acquiring only a part of light. Figure 4 shows schematic profile of unscattered beam I_u and intensity shift I_s at BFP. Unscattered light I_u has Gaussian distribution and intensity shift I_s has sine curve like distribution. When a particle moves along x axis only amplitude of sine curve changes in proportion to the particle displacement. At a center area in the BFP intensity of Gaussian beam is dominant and that of interference is weak. However interference is relatively stronger at the surrounding area. Thus intensity change at the surrounding area is more sensitive to particle position than the center area. Then we proposed an improved method BFP interferometry cutting off light at the center area. It corresponds to acquiring light at only sensitive surrounding area in BFP and higher sensitivity is expected by only putting a simple mask in the middle of BFP.

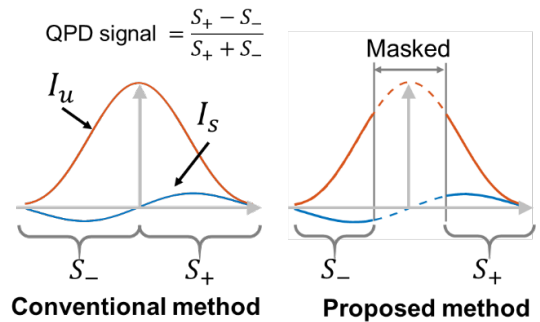


FIGURE 4. Schematic profile of intensity distribution

THEORETICAL ANALYSIS AND DISCUSSION

An intensity distribution and QPD signal is analyzed to investigate how masked area enhance a measurement sensitivity. A size of masked area is calculated as changing limits of integration ($\theta_{min} < \theta < \theta_{max}$) of Eq. (7). Table 1 shows simulation conditions. Under these conditions beam waist w_0 is about 260 nm. Therefore a range of particle displacement is defined to ± 300 nm range.

First to qualitatively analyze the proposed method intensity distribution with and without a mask are compared. Figure 5 shows intensity distribution as to displacement x (0, 30, 60 nm) of each

distributions: I_u , I_s , $I_u + I_s$ without a mask, and $I_u + I_s$ with a mask. Horizontal axis is along the x axis. The mask covers 60% of beam diameter. Because of verification analysis particle is relatively large 100 nm in diameter.

TABLE 1. Simulation conditions

Particle	Material	TiO ₂
	Refractive index n	2.9
	Diameter d (nm)	100
Laser	Wavelength λ (nm)	405
	Beam diameter w_{in} (mm)	3.2
Objective lens	Focus length f (mm)	1.8
	NA	0.93

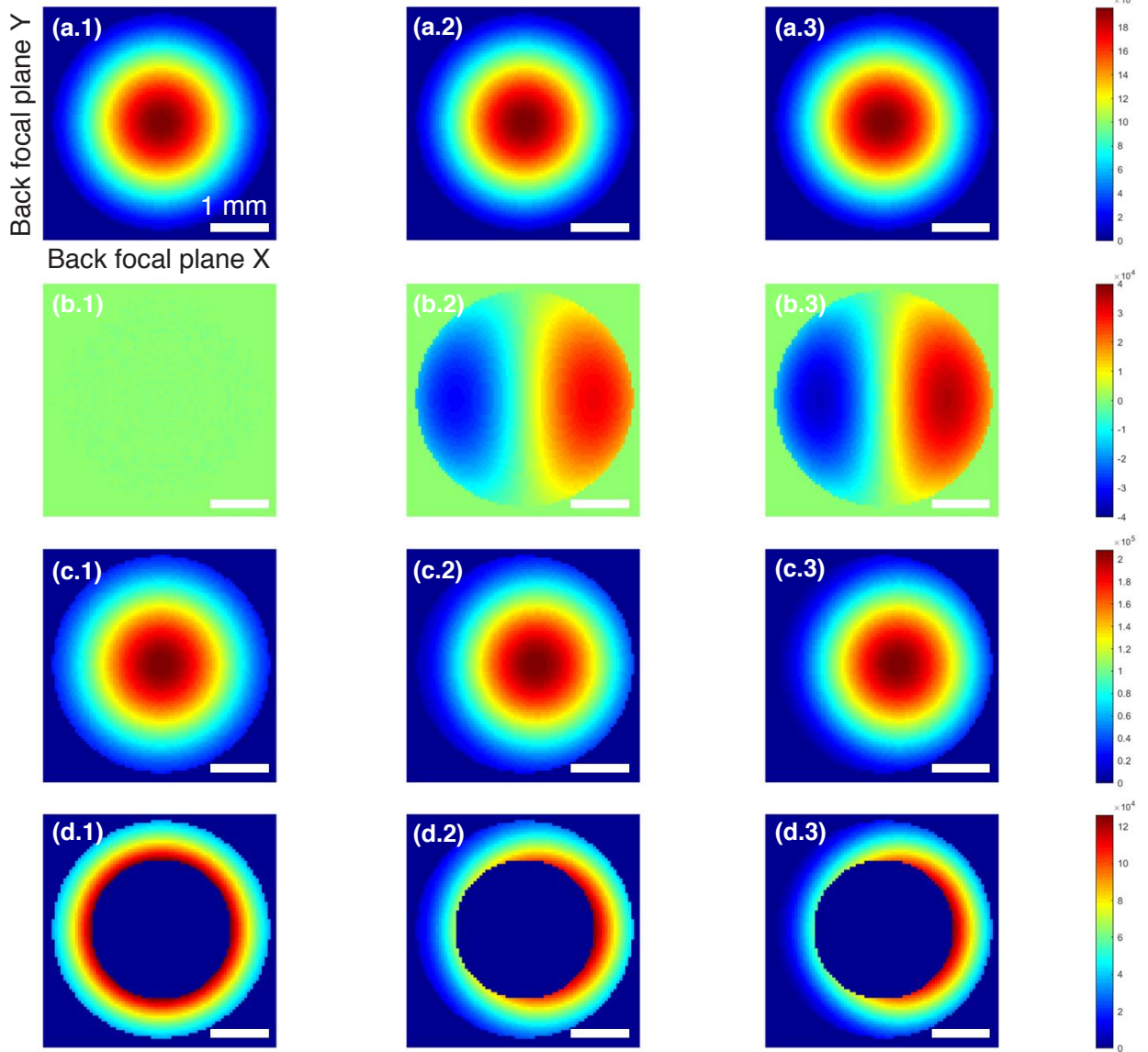


FIGURE 5. Intensity distribution in BFP (a combination of conditions (x, n) are follows:
a: Unscattered beam (I_u), **b:** intensity shift (I_s), **c:** $I_u + I_s$ without a mask, **d:** $I_u + I_s$ with a mask
1: displacement $x = 0$ nm, **2:** $x = 30$ nm, **3:** $x = 60$ nm)

Figure 5 (b.1)-(b.3) show equal and opposite semicircle distribution of the interference and a peak value of the intensity increases in proportion to a particle displacement. However when the displacement exceed a certain detection range, which is almost half of beam waist, the intensity decrease. Although in this report only one dimensional displacement x is analyzed, for two dimensional displacement this method can be applicable. Because unscattered light and intensity shift ($I_u + I_s$) have rotational symmetry distribution. Consequently the proposed method (Fig 5 (d.1)-(d.3)) provide clearer change than conventional one (Fig 5 (c.1)-(c.3)) and this change indicates higher measurement sensitivity. Figure 6 shows calculated QPD signals corresponding to displacement $x = \pm 300$ nm. In the range $x = \pm 70$ nm the almost liner signal allows us to position detection. Rapid variation, observed with larger masked area, means that position detection can be clearly improved by the proposed method. In conventional BFP interferometry high sensitivity reduces a detection range, which corresponds to peak-to-peak distance (Figure 6), because high sensitivity needs short wavelength and high NA. However the proposed method improves the sensitivity without decreasing a detection range. For quantitative comparison sensitivity β was defined as a peak-to-peak inclination. Figure 7 shows sensitivity β and light intensity against the size of masked area. Indeed acquired light amount decreases, sensitivity β is remarkably improved depending on masked area. For example, comparing 0% and 90 % about 4 times sensitivity β is achieved with obtained intensity of 3.0 %. Whereas with higher power of incident light a decrease in light amount at QPD is not a problem. In this report the conditions are fixed to only Table 1. However another condition may achieve higher improvement and sensitivity.

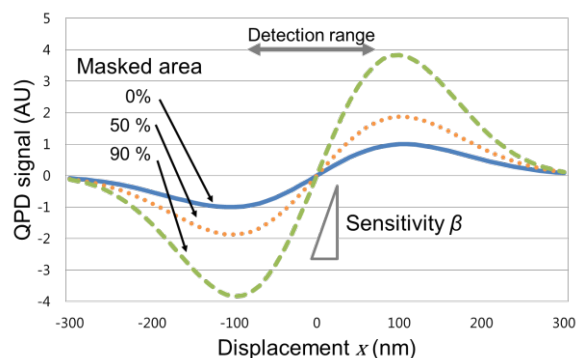


FIGURE 6. QPD signal against masked area

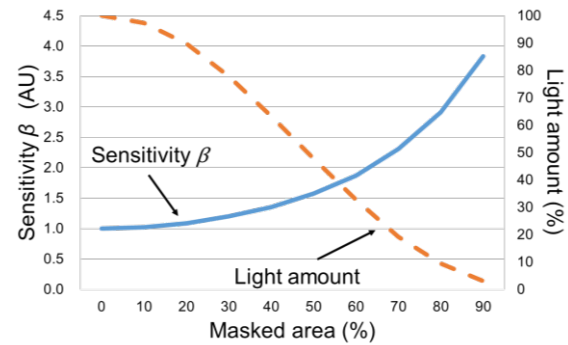


FIGURE 7. Relationship between masked area and sensitivity

CONCLUSION

We proposed improved BFP interferometry using a mask, which allows us to obtain a part of detection beam, and analyzed its characteristics. The theoretical analysis indicated the 4 times sensitivity. As a result it is confirmed sensitivity of position detection was clearly increased depending on masked area. Further work is needed to conduct a verification experiment.

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Scope of Metrology Systems

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METROLOGY IN PRECISION ENGINEERING

If an operational description of the discipline of engineering is

“taking the elements of nature and arranging them in such a suitable way as to benefit from that arrangement”

then **precision engineering** is

*“taking the elements of nature and their diverse assemblages with their law-abiding deterministic, observed, **measured**, analyzed, modeled, and controlled phenomenological properties, and **rationally arranging them in such a suitable way as to specifically, accurately, and optimally benefit from that arrangement.**”*

Measurement is then that *seminal precision engineering endeavor that makes possible a rational optimization* of an engineered benefit. McKeown described precision engineering as a “multi-disciplinary range of technologies, **based heavily on the application of metrology** to manufacturing” (quoted in [1]).

Without metrology there can be no precision engineering. You may be able to make a performance class violin with the skill of Stradivari but you will not know why it sounds the way that it does and be **rationally able to improve** it based on the properties of its elements and their arrangement. Perhaps parametric analysis based design is the reason why modern violins were qualitatively perceived to out-perform the classic Stradivarius class of instruments in evaluations involving performing artists [2].

METROLOGY AND MEASUREMENT SYSTEMS

Describing the necessity for a journal devoted to scientific instruments, Rayner states that “progress in science is **founded upon the results of measurements**, whether carried out primarily for the increase of natural knowledge,

or for the improvement of technical processes and apparatus used in all branches of applied science [3].”

Metrology is the encompassing “science of measurement and its application”. Any application of metrology requires **measuring system(s)**: “a set of one or more measuring instruments and often other devices **to give information used to generate measured quantity values...**”[4] These quantity values numerically describe the generic and individually distinguished properties of elements and their diverse arrangements including other precision engineered systems.

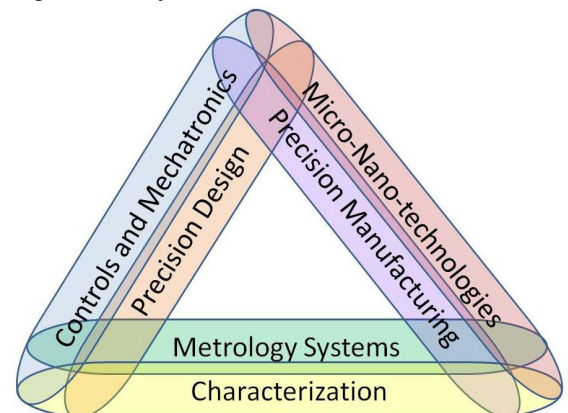


FIGURE 1. Technical leadership committee areas of the ASPE and their codependency.

Rayner went on to further emphasize, that if measurement is done without a proper knowledge, then the “apparent accuracy or sensitivity may be **entirely fallacious** even in the case of the most conscientious worker.” The aforementioned journal not only dealt with the design of the instruments but included a laboratory and workshop section on the manufacturing arts revealing that a metrology system does not exist in a vacuum, *i.e.*, separated from other aspects of precision engineering. They are codependent entities as illustrated in **FIGURE 1**.

Metrology systems are dependent on:

- 1) **Characterization** disciplines to assure that the quantity values provided by them are within their stated uncertainties,
- 2) **Controls and Mechatronics** disciplines to provide the technologies for quickly and accurately positioning sensors to provide related indications at specified locations as well as transducer optimization,
- 3) **Precision Design** disciplines to optimize the operational capability of the total system,
- 4) **Precision Manufacturing** disciplines to physically realize designed specifications that enable measurement, and
- 5) **Micro-nano-technologies** disciplines to decrease the obtainable sizes and increase capabilities of the sensors and related systems.

The general **purpose of a metrology system** is **to provide information toward obtaining the best estimate of the quantity value** of a properly defined measurand. The measurand can have a single base unit (dimension) of length, mass, time, temperature, etc or have higher dimensions associated with pressure, force, compliance, power, thermal expansion coefficient, etc. It may be that multiple indications from different instruments are necessary to obtain quantity values. Even the sensitivities of the measurement sensors themselves are quantified through other instrumentation.

Measurement realization path

Measurand → sensor → transducer → analyzer
→ indication(s) → combination → quantity value

As an example identifying the practical divisions of a metrology system, let the temperature of the air be the measurand. Let a material whose electrical resistance changes depending on temperature be placed within the environment as the sensor. The sensor is connected within an electronic circuit whose output value in volts changes as the resistance changes with temperature. This is the 1st transducer stage. The 2nd transducer stage converts the voltage value to a digital number to be used for computational analysis. The analysis uses a model that relates response of the sensor to the changes in environmental temperature. The calculated value from the model is then given as

the temperature in the form of a number value and an associated unit such as °C, °F, °R or K. The number value and unit is called the indication which can also be the quantity value. If indications from several sensors are used to report the average temperature or there are corrections that must be made based on other factors, then the final determined value is the quantity value for the measurand.

Diverse quantity values can be determined from a given set of indications. Consider a metrology system that provides information on a surface of a production part that is desired to be geometrically 'planar'. The measuring system can provide thousands of indications that relate the three dimensional spatial coordinates of that surface. These same indications can be used to determine quantity values for a myriad of defined measurands (parameters) related to the geometric planarity such as flatness [5], straightness, roughness, skewness, kurtosis, peak to valley etc [6]. These indications also can be useful for providing an analysis basis such as a bearing ratio curve or Q-Q plot etc. The methods for mathematically calculating these are often codified, for example, in written industry standards [5-9] while others are proprietary to a particular manufacturer. The defining of the parameters themselves could be considered a part of characterization disciplines while the indications and actual programmed calculations can be considered a part of the metrology system.

POEMS

The acronym POEMS divides the engineering disciplines that are involved in developing precision systems in such a way that they can be individually assessed.

Programmed
Optical
Electrical
Mechanical
Systems

As a subset, many metrology systems can also be considered as POEMS. Each incorporated subdivision requires rational optimization and quantification according to their respective disciplines to reduce the measurement

uncertainty and to extend the capability of such measurement systems.

Examples of capability differences are shown for the caliper systems in *FIGURE 2*. A simple mechanical caliper is shown measuring a valve component in *FIGURE 2a*. The effective vernier scale resolution is about 20 micrometers and requires visual reading. The dial caliper in *FIGURE 2b* has a mechanical transducer which provides an analog needle scale with a visual resolution of about 10 micrometers. The more advanced digital instrument of *FIGURE 2c*, also with a resolution of 10 micrometers has programming which allows switching between Imperial and Metric units and doesn't require a visual reading estimate. It has an internal electrostatic capacitance system as a position transducer, an optical LCD system for showing the quantity value, and a digital data output interface so that it can be incorporated in a system such as for statistical process control. Although more capable, the digital instrument also has an increase in cost over the less expensive vernier caliper. The digital system is a design requiring technology from all POEMS disciplines. The micrometer (Gascoigne) is a 1670's screw based improvement on Vernier's 1630's device and was further improved structurally demonstrating Abbe's principle.

The following questions reveal the relevance of advances in each discipline to the operation of a metrology system:

How is the **programming** effective and efficient in obtaining quantity values, extending the capability, controlling the positioning and accelerations of sensors, calculation of parameters from minimal indications to point clouds in accordance with standardized parameters, correcting indications for measurement conditions according to current and novel models, etc?

How do the **optical** subsystems enable the determination of the quantity desired through interferometric and other optical delivery arrangements, and wavelength transmission, absorption, penetration and reflection, efficiency, miniaturization, absolute and differential setups etc?

How do the **electronics** effectively and reliably act as transducers of the measured quantity? Are system and intrinsic noise-related limitations

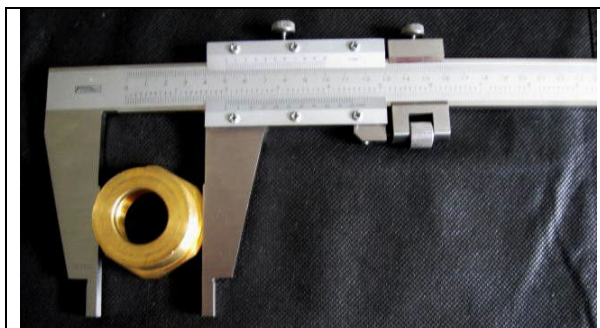


FIGURE 2a. Vernier calipers, mechanical measurement with vernier resolution of about 20 micrometers.

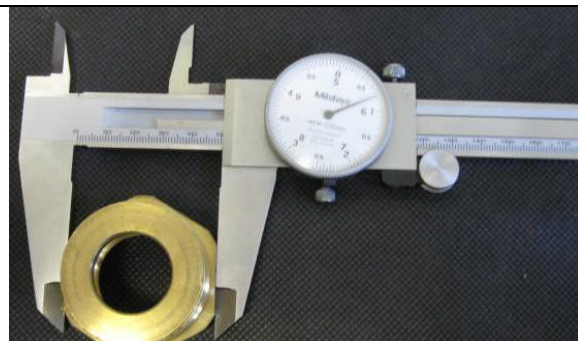


FIGURE 2b. Dial Calipers, mechanical transducer measurement with indicator resolution of about 10 micrometers



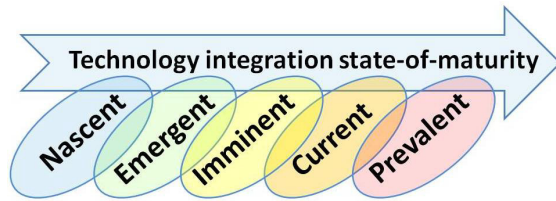
FIGURE 2c. Electronic calipers with electronic transducer and LCD optical readout. Resolution shown of 10 micrometers. Upper right corner of the display unit shows an external digital interface.

handled optimally, including response delay time between input quantity change and output values?

How does the **mechanical** system provide the framework necessary for measurement with minimal static and dynamic response to environmental conditions including temperature, humidity, pressure, radiation, illumination, vibrations, etc?

INTEGRATION MATURITY

The state-of-maturity for a particular technology is illustrated below:



Nascent technologies exist conceptually and may be theoretically demonstrated. **Emergent** technologies have been physically demonstrated in a research and development environment. **Imminent** technologies have been integrated into a prototype working system. **Current** technologies are those that are available to the open market. **Prevalent** technologies are the preferred choice in a particular application area. For example linear scale encoders for displacement measurement are prevalent on diamond turning machines while laser interferometry systems are still prevalent for microelectronic lithography machines. Current but yet-to-be-prevalent technology examples are X-ray computed tomography metrology systems.

ENCODER TECHNOLOGY STATE

For the precision engineer an encoder is a patterned (grating e.g.) based transducer that converts linear or angular displacement into an analog quadrature signal (binary or sine/cosine) for feedback and/or digital decoding.

The oral session that is aligned with this abstract has four papers concerning emergent and imminent encoder advances [10-13]. These are in bold italics in *Table 1*.

Current-maturity encoder capabilities include scale resolution of one nanometer [14,15, 50], start-up absolute position encoding [16,17], diagnostic and programming interfaces [18, 19], two-dimensional encoding [20], and carriage cementable scale tapes/ribbons [21, 22, 23, 49].

Emergent and imminent developments are shown in *Table 1* with italic grey/blue text (this conference) and smaller font for other encoder references. Multi-dimensional encoding is a major area of current investigation.

Table 1. Metrology systems recent developments. **Italic bold reference: encoder papers in this oral session, italic papers at this conference, small black: other recent encoder references.**

▼ Area ▼ For technical definitions see the VIM [4]	▼ State-of-the-art Metrology System Technology ▼	
	Emergent, Imminent	Current, Prevalent (encoder)
1. Measurand capability, novel systems	[10] heterodyne [26, 32, 33] [39,43,44,45, 46,47,48]	[16,17] absolute position [20] two-dimensional encoding
2. Measurement principle applications,	[10] interferometry [13] planar, [32, 36]	
3. Ranges, resolutions and their ratio,		[14,15,50] one nm resolution [21,22,49] ext. length scale tape
4. Process integrated measurement	[11] roll to roll	
5. Reducing the operator influence		
6. Selectivity, reducing non-measurand influence		
7. Stabilizing the structural/metrology frame	[42] [48]	[21] expanding scale tape
8. Parametric modeling		
9. Dynamic response	[30]	
10. Sensitivity		
11. Precision, repeatability, reproducibility	[35, 37] [41]	
12. Reliability		
13. Metrology methods and programming,	[13] 2D encoding [25, 26, 28] [40]	[18,19] diagnostics, interfaces
14. System models	[41]	
15. Noise reduction	[40]	[varied] 10 nm
16. Error, bias estimation and elimination	[12] harmonic error [28, 29, 31] [39, 40,48]	
17. Standards, uncertainty, and applications	[24, 27, 34, 38]	
18. Enterprise incorporation, IoT data	[27]	

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CONCEPTS AND GEOMETRIES FOR THE NEXT GENERATION OF PRECISION HETERODYNE OPTICAL ENCODERS

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A REVOLUTION IN STAGE METROLOGY

The fabrication of integrated circuits and solid-state memory relies on lithographic printing of overlaying patterns onto a semiconductor wafer. The manufacturing process may consist of more than 40 cycles, with each cycle defining a specific layer. Lithographic exposures require stage motions during a continuous projection, with the requirement that new patterns are registered with previous layers to much less than the minimum feature size or critical dimension (CD) of individual transistors, currently at 16 nm [1]. Double-exposure techniques permit an overlay error of no more than 15% of the CD, with about 20% of this number or 3% of the CD or 0.5 nm allocated for stage metrology [2].

As if the precision targets were not enough of a challenge, an additional requirement is stage speeds of greater than 1m/s, so as to achieve throughputs of 20s per wafer [3]. The wafer stage is continuously adjusted for tip and tilt through the exposure. These factors require closed-loop control with metrology feedback that continuously tests the limits of what is achievable.

A decade ago, stage metrology subsystems for photolithography equipment consisted entirely of line-of-sight laser interferometers. These systems, which still play a dominant role in stage positioning, are highly-evolved versions of Michelson two-beam interferometers configured to monitor displacements and changes in stage orientation using heterodyne detection [4].

As CD values have decreased, error analyses have revealed that 80% or more of the total error can be related to air fluctuation of the interferometer [1]. Serious attempts have been made to compensate for air-index using dispersion interferometry [4-6]; but these efforts have been overtaken by ever-tightening requirements.

Recognizing that optical encoders minimize the air turbulence contribution by reducing the air

paths, the photolithography industry now uses high-performance encoders to monitor the critical stage motions during wafer exposure [3, 7-11].

DESIGN CONSIDERATIONS

For complete motion metrology, a 2-D grating replaces the traditional stage mirrors, and the encoder system is expected to monitor all six degrees of freedom. The multi-axis motion monitoring is not only a basic requirement for advanced stage control, but is essential for the calculation and compensation of Abbe error. Consequently, the basic geometry and ray paths within each individual encoder must allow for dynamic changes in tip, tilt and distance to the read head during data acquisition, with acceptable signal loss, systematic error and cross coupling between measurement axes. Assuming that this can be done, there remains the task of designing encoders with sub-nm precision at data rates in the MHz regime.

Traditional optical encoders familiar to precision engineers employ homodyne detection and measure the in-plane relative motion between the encoder head and a metal or glass scale [12, 13]. Linear and rotary encoders can be found in a wide range of machine tools and precision instruments. They can be viewed either as interferometric devices, with the two interfering beams generated by diffraction; or as optical systems that project an image of the grating onto a detector. The basic resolution unit is given by the spacing between the grating lines, with higher precision achieved by interpolation.

Traditional encoders vary from inexpensive to high performance; but typical systems that one can buy off the shelf have performance characteristics that fall well short of the demanding requirements of modern photolithography stage positioning systems. Hence the need in recent years to develop new encoders, including the invention of fundamentally new geometries.

NEW DESIGNS

A way to solve the problem of tip/tilt sensitivity is to redirect a measurement beam twice to the encoder grating, in a strategy analogous to the double-pass plane mirror interferometer [4]. This can be achieved by modifying a retro-reflector based linear encoder with the grating placed in the beam path. FIGURE 1 illustrates an efficient and compact approach that uses the encoder grating to fold a linear interferometer back onto itself, resulting in sensitivity to both in-plane and out-of-plane motions [14]. The double pass with a retroreflection compensates for tip and tilt of the encoder grating.

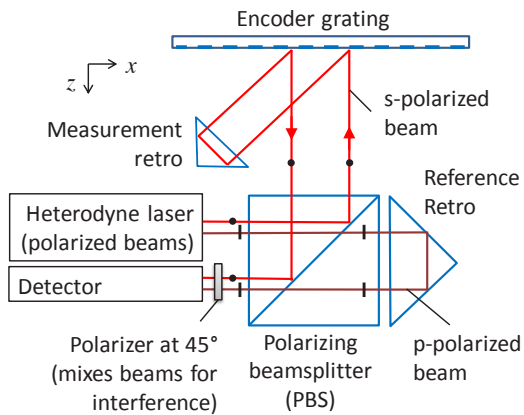


FIGURE 1: Basic encoder geometry using a linear interferometer.

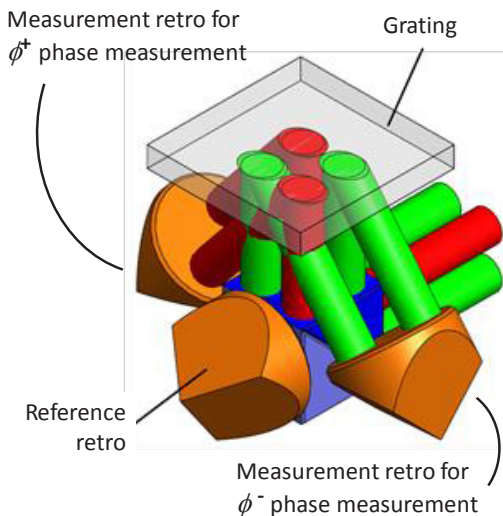


FIGURE 2: A two-axis encoder based on the concept shown in FIGURE 1.

Given the low diffraction efficiency of 2D gratings, signal to noise requirements point to a preference

for heterodyne interferometry, which benefits from coherent amplification and a measurement frequency offset to frequencies above thermal noise. The optics are fed with orthogonally polarized measurement and reference beams having a heterodyne frequency difference of 20 to 80MHz [15]. A polarizing beam splitter directs the light along reference and measurement paths, and a polarizer mixes the recombined beams at the detector.

Although the folded linear interferometer is compact and efficient, the in-plane x and out of plane z measurements are confounded in this elemental geometry. The solution is to employ two encoders sharing a common entrance beam and beamsplitter, providing two simultaneous phase measurements, ϕ^+, ϕ^- , as in FIGURE 2. The displacement measurements follow from straightforward math, where Λ is the grating pitch:

$$x = \frac{(\phi^+ - \phi^-)}{8\pi} \Lambda$$

$$z = \frac{(\phi^+ + \phi^-)}{8\pi} \frac{\lambda}{1 + \sqrt{1 - (\lambda/\Lambda)^2}}.$$

An important consideration in polarized interferometer design is the control of beam mixing. The cyclic error commonly encountered with interferometric displacement sensors can easily surpass several nm even with the best PBS optics [16]. To suppress these errors, angled beam geometries such as the one shown in FIGURE 3 combine with advanced firmware for the suppression of residual errors [17, 18].

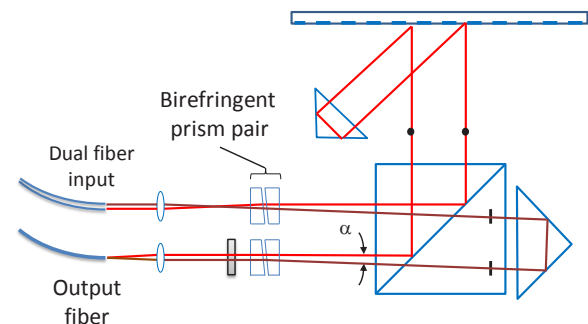


FIGURE 3: A technique for reduced polarization mixing uses separated-beam light delivery and angled beams.

Heterodyne encoders based on the folded linear interferometer solve the basic problem of tip/tilt compensation, but they have their limitations. Out-of-plane motion along the z axis has an undesirable side effect of laterally shearing the measurement beam with respect to the reference beam, which reduces signal strength. The result is a restriction in the allowable z displacement, which can be an issue if the metrology system must allow for a large range of motion in this direction.

The recirculating double-pass geometry shown in FIGURE 4 compensates for beam shear as the encoder head's distance to the grating changes [19]. After a first diffraction from the moving grating on the stage, the measurement beam diffracts from a fixed grating and reflects from a small roof prism back to the moving grating. A second cycle of diffractions from the two gratings compensates not only for tip and tilt, but also for beam shear caused by stage motion in the z direction.

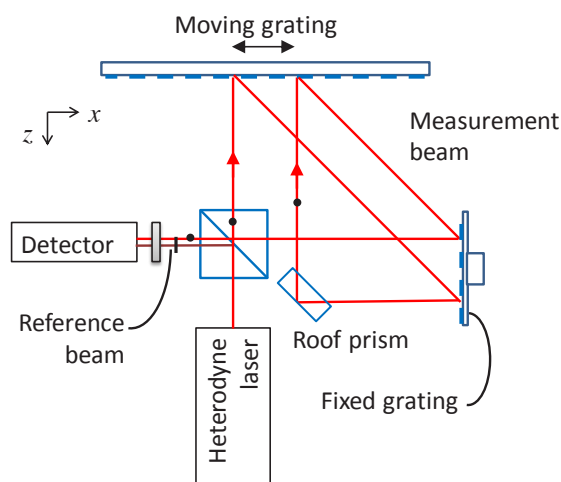


FIGURE 4: Simplified, conceptual drawing of a double-pass heterodyne encoder compensated for beam shear over large z -axis motions.

PERFORMANCE

The conceptual elements described in this abstract have been incorporated into advanced photolithography systems. In actual product designs, the optical components are rearranged so as to provide passive thermal compensation

as well as dimensional stability [20]. Many of these design details are currently proprietary, as are the procedures and results of system-level testing to verify the sub-nm uncertainty requirements of semiconductor lithography stage positioning.

Some pre-shipment testing is nonetheless an expected step in the manufacture of OEM encoder systems. The example spectral signal analysis in FIGURE 5 for a laser displacement interferometer shows the main signal as well as parasitic signals that are the result of polarization mixing, uncontrolled reflections, and other harmonic errors [4]. Tests of this kind performed for encoder heads in production confirm that cyclic errors are less than 0.05 nm RMS and the random noise is 0.06 nm for a 10 kHz bandwidth.

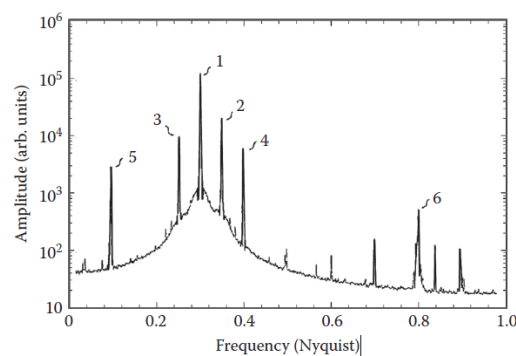


FIGURE 5: Graph showing the frequency content of a laser displacement interferometer signal. Data of this kind allows for the detection and correction of residual cyclic errors.

CONCLUSIONS

The demands of modern precision engineering exemplified by semiconductor photolithography systems have propelled encoder technology to a performance level surpassing that of line-of-site interferometry. The key advantage in these systems is the nearly complete suppression of air turbulence. This advance has come at the price of increased demands on calibration and compensation systems, as well as on the encoder design itself. Here we have reviewed the basic concepts and geometries underlying the current and future generation of optical encoders. Driven by ever decreasing critical dimensions, there will continue to be demands on developers for new designs with even higher precision.

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High precision interferometric optical encoder for inline position referencing of instrumented plastic film in a Roll-to-Roll system

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INTRODUCTION

High precision methods of measuring and controlling the alignment, motion and deformation of flexible substrates are required in order to scale advanced manufacturing processes for high-volume R2R production [1]. Example processes include precision thin film deposition, inkjet printing, nanostructure embossing, digital lithography, laser scribing, overlay and registered non-destructive inspection [2-5]. Such advanced manufacturing processes are critical in the supply chain of key enabling technologies.

The most common substrate position referencing strategy has been to infer the position and motion indirectly from roller rotation by employing rotary encoders. However, parasitic translation, slippage and deformation of the substrate material introduces significant differences between the inferred and actual film position, preventing truly high precision process control. Direct measurement is possible using machine vision systems e.g. edge tracking and imaging of printed fiducials [5-7]. The state of the art in machine vision based techniques applied in a R2R setting has achieved $2\text{ }\mu\text{m}$ accuracy in position measurement [8]. However, area scan imaging suffers from very low frame rates, on the order of 10^2 Hz. This limits its application for processes that require high frequency position information.

We present the concept and realisation of a high precision interferometric optical encoder with a high sampling rate, optimised to track the in-plane motion of plastic film that has been instrumented with continuous linear encoder scales in an R2R system (see FIGURE 1).

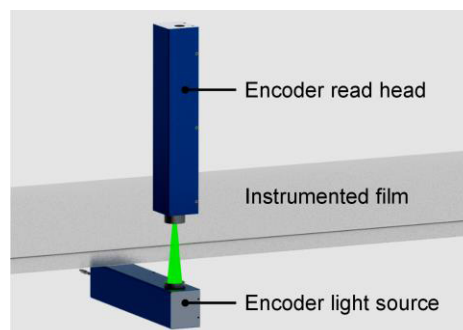


FIGURE 1. Encoder concept design.

DISPLACEMENT MEASUREMENT

Various schemes exist for optical encoders, ranging from the simple readout of a chopped optical signal to the readout of the interference signals generated by grating interferometers [9]. Most encoders are integrated with a rigid prefabricated encoder scale of proprietary design and finite length, often providing additional functionality such as absolute position and multiple degrees of freedom measurement, with the primary application of providing position feedback for high-precision mechatronic systems [10]. As such, the encoder scales are often non-trivial to fabricate. In order to minimise the complexity and cost of continuous encoder scale fabrication in the R2R system, an interferometric encoder scheme requiring a simple binary phase grating has been selected as it does not require a reflective coating and is easily implemented [11]. In this scheme, displacement measurement is achieved by projecting coherent light through the encoder scale, such that the diffracted light interferes on an array of detectors, and measuring the phase of the resulting fringe pattern.

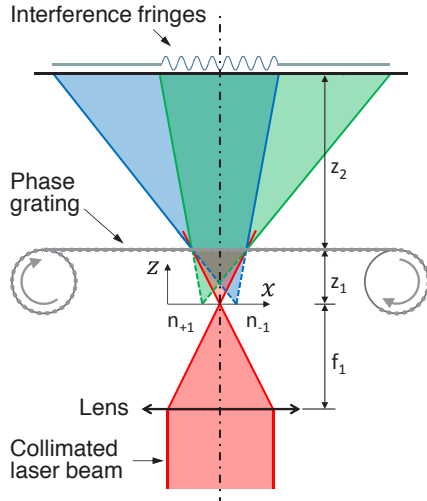


FIGURE 2. Schema of the optical system.

In the schema shown in FIGURE 2 a collimated laser beam is focused and then projected through a zero order suppressing phase grating. Zero order suppression can be achieved by a square profile grating with 50 % duty cycle when its amplitude satisfies the condition for destructive interference of light passing through the upper and lower interfaces at the operating wavelength.

$$a_m = \frac{m\lambda}{2(N_2 - N_1)}, \quad (1)$$

where m is any non-zero integer, λ is the wavelength of the laser and N_i are the refractive indices of the grating material system. Monochromatic light passing through the grating is diffracted, producing a series of diverging beams that appear to originate from a unique virtual focal point, one for each diffraction order. For small diffraction angles, the lateral position of these focal points in the focal plane of the lens can be estimated from the diffraction equation as:

$$x_n = -z_1 \tan(\sin^{-1}(n\lambda/p)), \quad (2)$$

where z_1 is the displacement of the grating from the lens' focal plane, n is the selected diffraction order and p is the period of the phase grating. Here, the sign convention of the diffraction orders is defined by the sign of the additional in-plane k-vector given to the light by the diffraction. The phase of light scattered into each diffraction order advances or recedes in proportion to the displacement of the grating according to:

$$\varphi_n = 2\pi n\Delta x/p, \quad (3)$$

where Δx is the in-plane displacement of the phase grating perpendicular to the grooves. As such, the $\pm 1^{\text{st}}$ diffraction orders experience opposite phase shifts. When allowed to propagate and interfere on a detector a fringe pattern will be observed according to the general equation that describes the interference of two point sources of coherent light:

$$I(x, y, z) = \frac{A_1^2}{r_1^2} + \frac{A_2^2}{r_2^2} + 2 \frac{A_1 A_2}{r_1 r_2} \dots \cos(k_1 r_1 - k_2 r_2 - \varphi_1 + \varphi_2), \quad (4)$$

where A , φ and k are the amplitude, phase and wavenumber of the sources, respectively and $r_i = \sqrt{(x - x_i)^2 + (y - y_i)^2 + (z - z_i)^2}$ is the distance between detector coordinates (x, y, z) and each source's coordinates (x_i, y_i, z_i) . In the plane defined by $(x, y, z = \text{constant})$ the fringes follow the characteristic arcs of hyperbolae, see FIGURE 3. It is clear that the phase of this fringe pattern depends on the initial phase of the sources, which are in turn related to the displacement of the grating via equation 3.

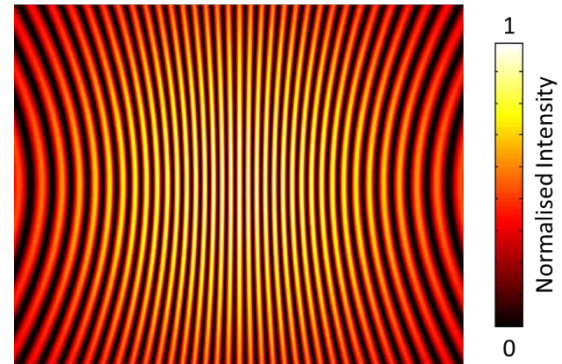


FIGURE 3. Image of the characteristic fringe pattern produced by the interference of two point sources of coherent light in the detector plane $(x, y, z = \text{constant})$. Source parameters have been adjusted to exaggerate the hyperbolic form of the fringe pattern.

A simple fringe phase determination algorithm can be applied by sampling this pattern in ideal quadrature *i.e.* measuring the interference signal at coordinates with phase shifts of exactly 0, 90, 180 and 270 degrees to obtain the sine and cosine signals from which a Lissajous figure can be plotted and phase determined using *via*:

$$\varphi = \tan^{-1} \left(\frac{I_A - I_C}{I_B - I_D} \right) \quad (5)$$

Finally, using this phase, displacement can be found according to:

$$x = \varphi \frac{p}{|n_1 - n_2|}, \quad (6)$$

where n_i are the signed integer values of the selected diffraction orders.

While sampling with four point detectors is possible, it can result in poor signal to noise ratio since most of the fringe pattern goes unused. An area or line scan camera can sample the fringe pattern at many thousands of points in parallel, however since the pixels on such sensors are rigidly arranged at a fixed pitch this leads to non-ideal sampling of the fringe pattern for the above fringe phase determination method [12]. By adjusting the geometry of the optical system, the distribution of the fringes can be tailored such that they appear parallel and with close to a single frequency over the sensing region of the camera. By tuning this frequency to be one quarter of the spatial frequency of the camera's pixels (inverse of pitch), the ideal quadrature condition can then be satisfied.

Non-ideal quadrature can result from errors in the grating geometry, the laser source's wavelength, and nonlinearities in the sensing opto-electronics or the mechanical alignment of the encoder with respect to the web. For example, a displacement of the grating along the optical axis of the encoder towards the focal plane of the lens will decrease the frequency of the fringe pattern seen on the detector. Non-ideal quadrature introduces errors in the simple fringe phase determination method described above. Performance under non-ideal quadrature can be improved by applying a phase shifting algorithm (PSA) that is robust against detuning [13]. We have chosen to implement such PSA methods but they are outside the scope of this paper.

UNCERTAINTY

A formal uncertainty budget is currently under development; however, initial analysis indicates that the static and dynamic relative alignment of the encoder light source, instrumented substrate and encoder read head in six degrees-of-freedom are expected to contribute most significantly to the total uncertainty. A key source of changes in relative alignments is the dynamic behaviour of the web itself, in particular, vertical flutter. While the uncertainty contribution from such errors can be minimised, dynamic errors will continue to contribute to the uncertainty of the measurement.

Consequently, web-handling performance should be optimised. Alternatively, simultaneous measurement of additional degrees of freedom could be made in order to apply a correction.

REAL TIME SIGNAL PROCESSING

To enable measurement at process-relevant speeds in the R2R system, the encoder read head has been implemented with a line scan detector that can sample the interference fringe pattern at a rate of up to 80 kHz with sufficient signal to noise ratio. A PSA has been implemented directly on the same field-programmable gate array (FPGA) hardware as the frame grabber to enable close to real-time fringe phase analysis. Thus far, the FPGA implementation has been optimised to achieve a calculation rate of 50 kHz. For the intended encoder scale geometry fabricated by the R2R system (40 μm period), this rate corresponds to continuous position referencing at substrate velocities of up to 0.25 $\text{m}\cdot\text{s}^{-1}$ (or 15 $\text{m}\cdot\text{min}^{-1}$).

EXPERIMENTAL REALISATION

The proposed R2R system for fabrication of the encoder scales is shown in FIGURE 4. It shows the space available for the metrology module containing the optical encoders.

The encoder scales are formed by ultra-violet (UV) nanoimprint lithography as follows [14]:

1. The plastic substrate is unwound from the delivery module and slot-die coated with a photopolymer resist.
2. Continuous surface relief gratings in both the x- and y-axis are then mechanically embossed in to the resist from the master template, which has been diamond-turned in to the copper embossing drum.
3. The embossed resist is cured with a bank of UV lights before lift-off.

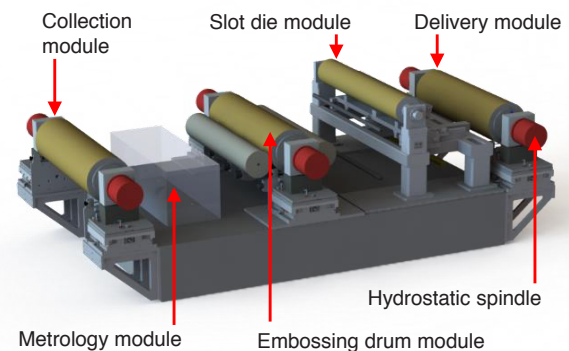


FIGURE 4. Schema of the system for R2R fabrication of gratings.

The encoder opto-mechanics are rigidly mounted on a machined flat. The optical path is nearly fully sealed within a tube system in order to minimize the effect of air disturbances on the fringe pattern and an external enclosure further isolates the system from the environment (FIGURE 5). The encoder systems are held in place by a gantry structure that spans above and below the instrumented web and a pair of high precision guiding rollers in close proximity minimises the unsupported span of the web to reduce the effect of web flutter.



FIGURE 5. Photo of the optical encoder.

Multiple encoder heads are to be deployed on the R2R platform in the space following the encoder scale lift off from the embossing drum (see FIGURE 6).



FIGURE 6. Embossing drum of the R2R system. The three lanes of x-y encoder scales are visible on the surface of the drum and the UV curing system can be seen beneath.

CHARACTERISATION

To verify the fringe pattern behaviour, the line-scan camera was temporarily replaced with an area scan camera. FIGURE 7 (a) shows a zoom of the instantaneous fringe pattern that is found at the detector plane. The phase of this fringe pattern shifts to the right or left depending

on the direction of motion following the description above. It was also possible to observe and correct a rotational misalignment of the fringe pattern with respect to the camera axes. FIGURE 7 (b) shows the Lissajous figure plotted from the linear motion of the grating, corrected for linear error in the grating pitch. FIGURE 7 (c) shows the displacement residuals obtained in the measurement of linear motion by direct comparison to reference displacement metrology. The standard deviation of the residuals in this measurement was found to be 32 nm, demonstrating the sub-micrometre performance of the system. A measurement noise of 27 nm was determined by taking the standard deviation of displacements sampled while holding the encoder scale in a stationary position for the same duration and at the same sampling rate as the linear motion test.

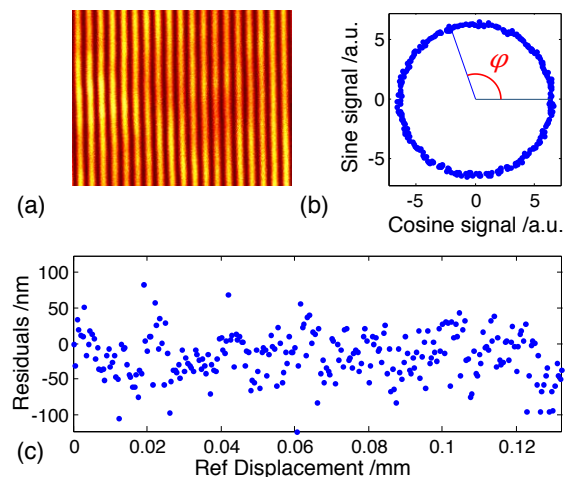


FIGURE 7. Measurement of linear web motion. Encoder is triggered by signals from a reference encoder every 500 nm. a) Fringe pattern at the detector. b) Lissajous figure. c) Displacement residuals versus reference encoder position ($\sigma = 32$ nm).

The system has also been characterised with the fast line-scan camera. At the achieved 50 kHz PSA calculation rate the maximum exposure time for the line-scan camera becomes very short ($\sim$ microseconds). In order to obtain sufficient signal to noise ratio the optical system has been optimised to maximise the intensity of the fringe pattern on the detector. The measurement performance with the line scan camera has been found to be equivalent or better than the area-scan implementation.

CONCLUSION

In conclusion, we have demonstrated an interferometric optical encoder with true sub-micrometre displacement measurement performance and a high sampling rate that is suitable for in-line position referencing of instrumented plastic film in a Roll-to-Roll system.

ACKNOWLEDGEMENTS

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AUTOMATIC HARMONIC ERROR DETECTION AND CORRECTION IN SINUSOIDAL ENCODERS

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ABSTRACT

The presence of higher harmonics in the signals of sinusoidal encoders is one of the chief remaining error sources in deriving position values from these encoders using high-performance interpolator circuits and algorithms. A new technique permits an interpolator to automatically detect the presence and magnitude of these harmonics and to correct for the position errors they introduce. This yields a significant reduction in the overall position measurement errors, and permits the use of more cost-effective encoders that have not gone through the expensive calibration typically required to minimize these harmonics.

INTRODUCTION

Sinusoidal encoders are becoming increasingly popular as the main position feedback devices in high-precision applications. Modern "interpolation" circuits and techniques can provide resolutions from these sensors at the nanometer level and even below.

However, this very high resolution does not necessarily yield comparable accuracy levels. Fundamentally, standard interpolator circuits expect perfect "sine" and "cosine" signals, computing an arctangent value to obtain high-resolution interpolated position within a single cycle of the signals. Imperfections in the signal, both repeatable and non-repeatable, will result in errors in the interpolated position from such a circuit. Even when these errors are small, they can lead to significant roughness in velocity of moves due to aliasing of the errors down to the servo update frequency.

It has been possible for some years now in sophisticated interpolators for the user to be able to enter correction factors for basic repeatable errors. But determining these correction factors has been an off-line process, often a tedious manual one, with different

corrections required for each individual encoder, even of the same model. And if there is a subsequent change in the signal errors, as with changing focus over distance or thermal drift, position errors are re-introduced.

A recent innovation employed very high sample-rate analog-to-digital converters and fast embedded digital logic to automatically detect fundamental signal errors and correct for them. This permits almost total elimination of the errors resulting from DC offsets in the sine and cosine signals, mismatch of the magnitudes of these signals, and phase error from ideal quadrature. Because these errors are detected and corrected in-line, no additional work is required of the user, and the corrections can change automatically as the errors vary over distance and time.

Now a new technique provides similar automatic detection of, and correction for, the effects of higher harmonics in the signals. In many cases, these will be one of the largest remaining sources of position errors in the system. Encoders that have very low higher-harmonic signal content are usually very expensive due to the time-consuming process the manufacturer must employ to minimize these signal errors. This new interpolation technique can permit the use of less expensive encoders that have significant higher harmonic content without compromising the resulting position accuracy.

BACKGROUND

The most prevalent "direct" interpolation technique for many years has been to use dual-rate sampling. A high-rate but low-resolution (1-bit) sampling circuit, typically operating in the megahertz range, provides the encoder line count information. A low-rate but high-resolution sampling circuit – typically employing 10 to 16-bit ADCs and operating in the kilohertz range – provides the sub-line, or interpolated, position information needed for each software servo

cycle. The block diagram for this technique is shown in Figure 1.

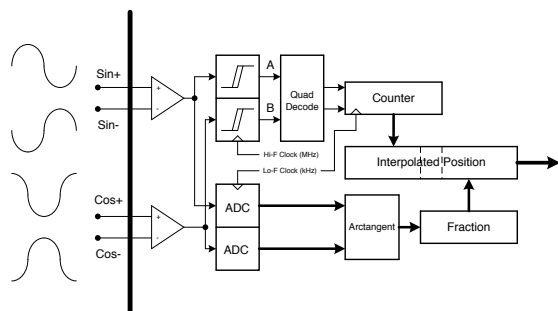


FIGURE 1. Traditional Dual-Sample-Rate Direct Interpolator

While the high-rate sampling circuit satisfies the Nyquist sampling criterion for the overall signals, its low resolution does not permit it to detect any error patterns in the signal. Analytically, the sinusoidal signals can be decomposed into “ideal” and “error” components, with the error components having the same and higher frequencies as the ideal components.

The high-resolution ADCs in a traditional interpolator, typically of the successive-approximation register (SAR) type, do not have the sample-rate capability to satisfy the Nyquist criterion for the error components that are at the encoder line frequency and above. Even though these error frequencies are far above the physical bandwidth of the system, the sub-Nyquist sampling often aliases these errors down to within the system bandwidth, yielding position errors and rough operation. A new technique is necessary to improve on this.

The recent introduction at reasonable prices of ADCs with both high resolution and high sample rates, plus the availability of high-density programmable logic devices (FPGAs) that can operate at these frequencies permit a new strategy to compensate for these errors. These ADCs are a hybrid of flash and SAR technologies, with the fast flash circuit providing most of the resolution, and a pipelined SAR stage to add several bits of resolution with minimal delay.

These technologies yield a new type of interpolator, with full-resolution sampling at above the Nyquist minimum frequency. The overview of this technique is shown in Figure 2.

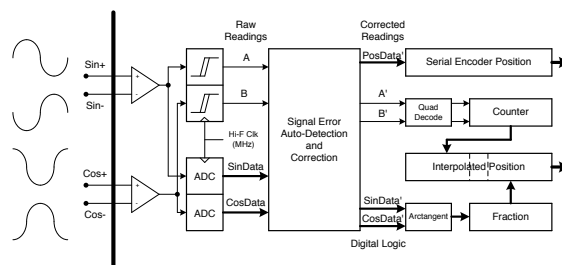


FIGURE 2. High-Sample-Rate Correcting Interpolator

FUNDAMENTAL-FREQUENCY SIGNAL ERRORS

The first errors detected and corrected using this style of interpolator were the “fundamental” signal errors: sine offset, cosine offset, magnitude mismatch between sine and cosine, and phase error. These signal errors are shown in Figure 3.

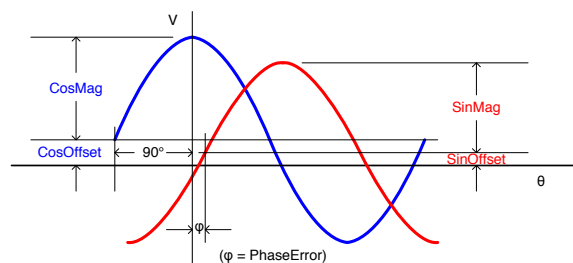


FIGURE 3. Sinusoidal Encoder Fundamental Signal Errors

The dominant position errors from each of these signal error types are independent and individually identifiable as shown here (where “x” is the fraction of the signal magnitude, “ ϕ ” is the offset from ideal quadrature:

- 1.) Sine voltage offset: $E = x \cos \theta$
- 2.) Cosine voltage offset: $E = -x \sin \theta$
- 3.) Magnitude mismatch error: $E = (x/2) \sin 2\theta$
- 4.) Phase offset error: $E = -\phi \cos 2\theta$

HIGHER-HARMONIC SIGNAL ERRORS

It is common for analog quadrature encoders, even those marketed as “sinusoidal” encoders, to have substantial odd-harmonic content, notably in the 3rd, 5th, 7th, and 9th harmonics. These signal harmonics produce predictable higher even-harmonic position errors in an interpolator circuit that assumes these signal components are not present.

A matching 3rd-harmonic component in both signals yields position errors of 4th, 8th, 12th, 16th etc. harmonics of the encoder cycle frequency in an interpolator that expects no higher signal harmonics. If this component is of fraction x of the fundamental signal component, the resulting position error (in radians of a line cycle) can be expressed as:

$$E = \frac{x^1}{1} \sin 4\theta + \frac{x^2}{2} \sin 8\theta + \frac{x^3}{3} \sin 12\theta + \frac{x^4}{4} \sin 16\theta + \dots$$

For example, with $x = 0.05$, the 4th-harmonic component of the position error has a magnitude of 0.05 radians, or 2.86 degrees, of a line cycle, and the 8th-harmonic component has a magnitude of 0.00125 radians, or 0.07 degrees. Figure 5 shows the signal waveforms (a), the resulting position errors (b), and the remaining position errors after the dominant component has been removed (c) for this case.

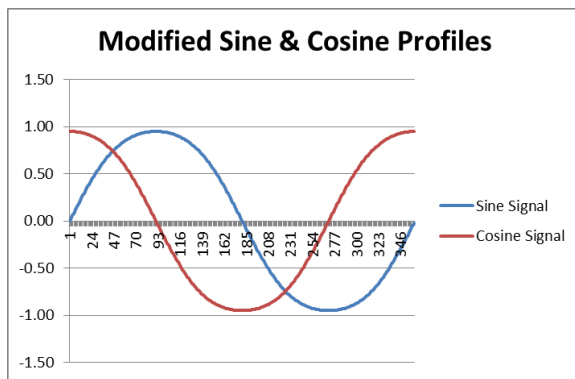


FIGURE 5a. Sinusoidal Encoder Signals with +5% 3rd-Harmonic Component

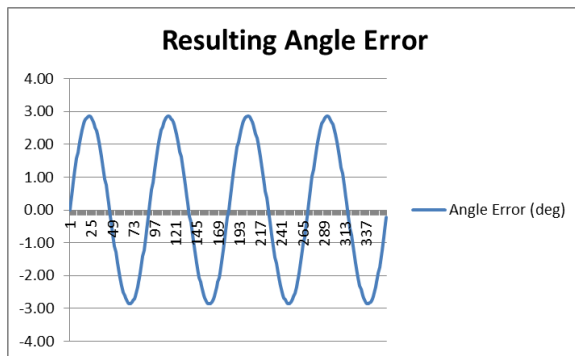


FIGURE 5b. Resulting Position Errors from +5% 3rd-Harmonic Component

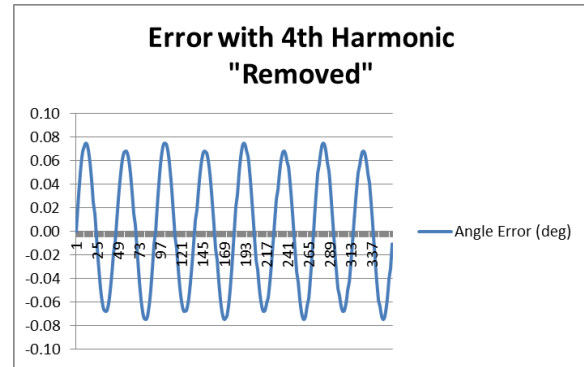


FIGURE 5c. Remaining Position Errors after Dominant Component Removed

Similarly, the position errors resulting from a matching 5th-harmonic signal component can be expressed as:

$$E = \frac{x^1}{1} \sin 4\theta - \frac{x^2}{2} \sin 8\theta + \frac{x^3}{3} \sin 12\theta - \frac{x^4}{4} \sin 16\theta + \dots$$

Matching 7th-harmonic signal components yield errors of:

$$E = \frac{x^1}{1} \sin 8\theta + \frac{x^2}{2} \sin 16\theta + \frac{x^3}{3} \sin 24\theta + \frac{x^4}{4} \sin 32\theta + \dots$$

Matching 9th-harmonic signal components yield errors of:

$$E = \frac{x^1}{1} \sin 8\theta - \frac{x^2}{2} \sin 16\theta + \frac{x^3}{3} \sin 24\theta - \frac{x^4}{4} \sin 32\theta + \dots$$

INTERACTIONS OF ERROR SOURCES

If the only signal errors are matching odd harmonics, the resulting position errors are independent and superimposable, with only $\sin(4N\theta)$ components. However, if there are fundamental signal errors as well, or the harmonic magnitudes are not the same between the sine and cosine signals, there can be cosine position error components as well, and with frequencies of $2N\theta$ as well. A robust algorithm must be able to detect and correct for these as well.

DETECTION AND CORRECTION METHOD

Figure 6 shows a block diagram of a method for automatically detecting and correcting the position errors resulting from these harmonic signal errors.

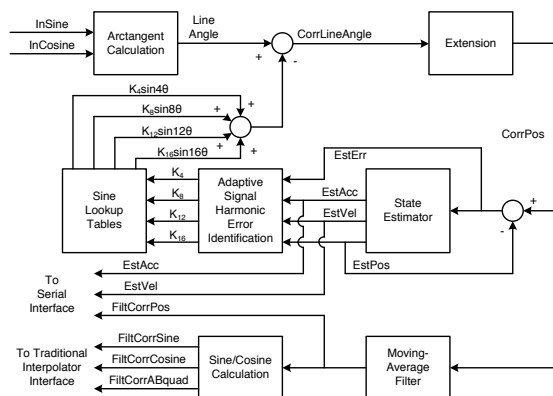


FIGURE 6. Block Diagram for Automatic Detection and Correction of Position Errors Resulting from Harmonic Signal Errors

The “Arctangent Calculation” block converts the digitized sine and cosine signals into an uncorrected angle within one line of the encoder. Considering first without corrections, this is extended to a multi-line position value by including the number of full sine/cosine cycles elapsed.

This position value is then fed into the digital “State Estimator” block, which contains a simplified model of the system. It fundamentally acts as a low pass filter. Given that the position errors produced by signal harmonics are at very high temporal frequencies at all but the lowest speeds, the operation of the estimator is not very sensitive to the values of the model constants chosen, especially if the identification algorithm is only activated above a minimum velocity threshold.

Fundamentally, the high-frequency errors between the input position and the estimator position can be attributed to measurement errors because the physical system cannot truly operate at these frequencies.

The error values are fed into the “Adaptive Signal Harmonic Error Identification” block, which looks for patterns in the errors. This figure shows a block that finds the sine components of the 4th, 8th, 12th, and 16th harmonics of the position errors, but other components are possible.

The identification is done through a Fourier transform algorithm. With only specific frequencies being examined, this is much simpler than a “continuous” transform. Figure 7

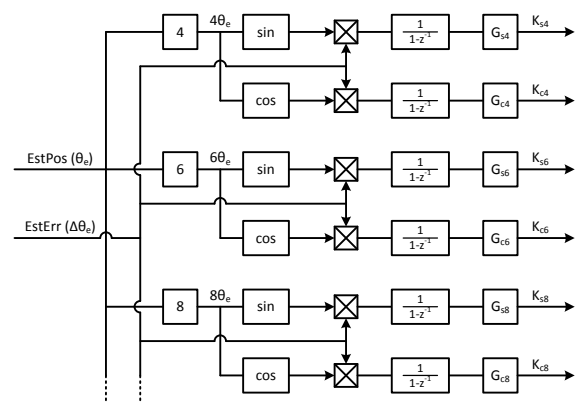
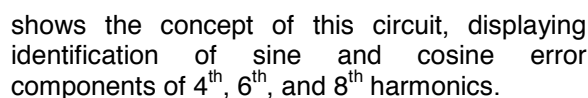


FIGURE 7. Error Identification Circuit

Each sample cycle, the error coefficients calculated in this block are multiplied by “Sine Lookup Table” values based on that cycle’s position. The resulting products are combined to obtain an overall instantaneous error value, and this is subtracted from the uncorrected position value.

With the error identification and correction algorithm always running (above a minimum velocity threshold), the interpolator can continuously adapt to error patterns that change over position and time. This permits high-accuracy results from encoders that are less expensive to make and to install.

CHOICE OF COMPONENTS TO REMOVE

The decision as to what and how many position-error components to remove is a classic engineering trade-off. Obviously, the error components that tend to be the largest are the most important to remove. The higher the harmonic in the signal, the lower the magnitude tends to be. For each signal-error component, the secondary, and especially tertiary, position-error components are significantly smaller than the primary. The cosine components from interactions tend to be smaller than the sine components.

Since each position-error component requires significant additional logic to detect and correct, the cost of this logic must be judged by its effectiveness. Typically, correcting for 10 components will reduce the resulting cyclic

position errors by about two orders of magnitude.

NON-REPEATABLE ERRORS

In addition to these types of repeatable signal errors, the accuracy of the position results is limited by non-repeatable measurement noise, whether from electromagnetic interference or digitizing quantization. Even though the much of the frequency content of this noise is beyond the physical bandwidth of the system, the sub-Nyquist sampling rate of traditional interpolators will alias this content to lower frequencies that will affect the physical system response.

The radical oversampling of this new circuit permits significant filtering to reduce the resulting variation without inducing noticeable time delay. Earlier approaches utilized a simple IIR exponential filter that did not require much logic or storage. With increased density and cost-effectiveness of programmable logic, FIR moving-average filters can be employed, which result in faster responses with equivalent filtering effect.

EXPERIMENTAL RESULTS

Figure 8 shows how the digital oversampling dramatically reduces the measurement noise in the resulting position on a mechanically locked stage. The non-oversampled position values show much higher noise levels (RMS $\sim 1 \mu\text{m}$) due both to quantization and EMI, than the oversampled and averaged ($\sim 0.1 \mu\text{m}$).

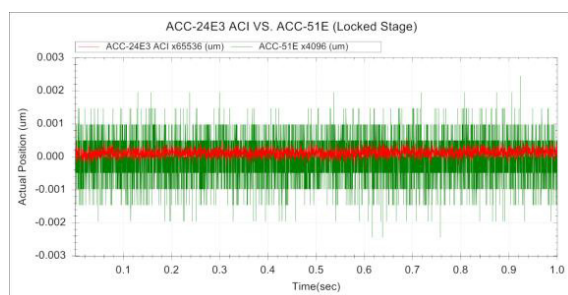


FIGURE 8. Oversampled vs. Non-Oversampled Position Measurements, Locked Stage

Figure 9 shows the performance of a system to a move with parabolic velocity profiles without the automatic detection and correction of repeatable signal errors. The high-frequency position (following) errors are primarily from these signal errors (oversampling is active to

reduce non-repeatable errors). Note that there is no integral gain active in the servo position loop.

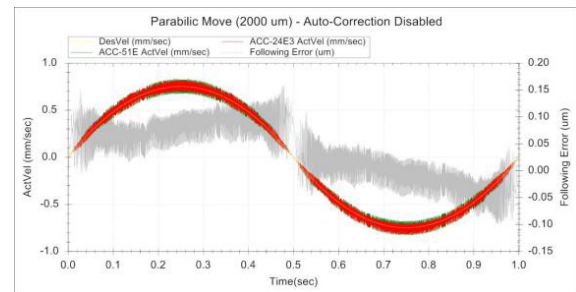


FIGURE 9. Parabolic Profile Move Without Automatic Correction

Figure 10 shows the same move on the same system, but with automatic detection and correction of repeatable signal errors enabled at 0.15 seconds – when the velocity magnitude exceeded a threshold. Note the dramatic reduction in actual velocity ripple and position error variation. The reductions are maintained at low velocities; even though correction values are not being updated, previously determined correction factors are still employed. Again, there is no integral gain active.

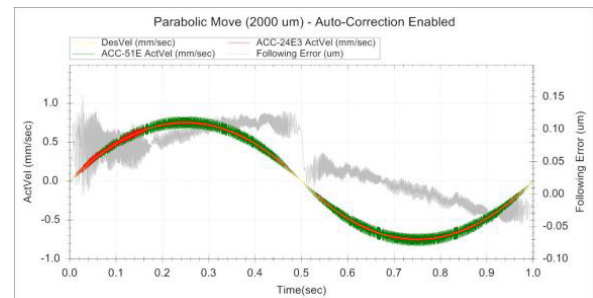


FIGURE 10. Parabolic Profile Move With Automatic Correction

USE OF ESTIMATOR VELOCITY AND ACCELERATION VALUES FOR SERVO FEEDBACK

In standard digital positioning servo systems, only an actual position value sampled once per servo cycle is used for feedback. Velocity values (and acceleration values, if used), whether explicitly calculated (as in derivative gains) or implicitly (as in a lead filter), rely on backward difference equations at the servo sample rate.

The differencing introduces increased quantization noise, and the “backward” differencing introduces additional phase lag, as

the velocity value is effectively one-half cycle old, and the acceleration value is a full cycle old.

However, the state estimator in the interpolator is continually computing velocity and acceleration values by numerical integration at a vastly (~1000 times) higher sample rate. The resulting values have lower quantization noise and less time lag. The use of these values for actual velocity and acceleration in the servo loop can yield improved damping and “electronic flywheel” effects.

CONCLUSION

The ability of a sinusoidal interpolator to automatically detect and correct for cyclic signal errors, including the presence of higher harmonics, brings unprecedented accuracy to the resulting interpolated position values. In addition (or alternatively), it permits the use of less expensive encoders that have not been finely and expensively calibrated to minimize these signal errors. The oversampling in these interpolators permits a substantial attenuation of measurement noise without the introduction of any significant delays. In addition, the intermediate acceleration and velocity states computed by the embedded state estimator can be used directly for servo feedback, providing values with lower quantization noise and less delay.

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A VISION-BASED ABSOLUTE PLANAR ENCODER FOR HIGH PRECISION IN-PLANE POSITION MEASUREMENT

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INSTRUCTION

This paper presents a novel vision-based planar encoder for absolute high precision three-degree-of-freedom ($XY\theta_z$) position measurement. This planar encoder is composed of a grating and a black-and-white (B/W) camera that captures the images of the grating. The grating has a periodic grid pattern with embedded codes for absolute position measurement. By decoding the embedded codes, absolute position measurement can be made with a resolution equal to the pitch of the grid pattern. For higher measurement resolution, a Discrete-Fourier-Transform (DFT)-based phase estimation method is used to determine the position within each period of the grid pattern. Experimental results show that the proposed planar encoder can potentially achieve a resolution similar to that of laser interferometers, but with a much simpler system.

PRINCIPLE

Figure 1 shows the configuration of the proposed planar encoder, which is composed of an 8-bit B/W camera with a $1\times$ low distortion lens. The grating is made of quartz glass with a layer of chrome coated on its surface to form the

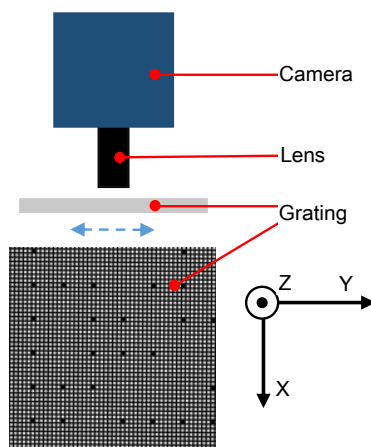


FIGURE 1. Configuration of the proposed planar encoder.

designed pattern. The camera captures the image of the grid pattern as the grating or the camera moves in the XY plane or rotates about Z-axis. The image is then processed to determine the relative position between the grating and the camera.

As shown in Figure 1, the grating is essentially a periodic grid pattern with embedded absolute position codes that are sparsely distributed. The periodic grid pattern is used to determine the position, denoted as (x_r, y_r) , of the image center relative to the grating within one grid period by a phase estimation method. The absolute position codes are formed by selectively removing designated grid squares from the grid pattern. By decoding these codes, the absolute position of the image center, denoted as (x_a, y_a) , can be obtained with a resolution equal to the grid period. The final high-resolution absolute position of the image center (x_c, y_c) is the sum of (x_r, y_r) and (x_a, y_a) . Figure 2 shows the flowchart of the image processing algorithm.

Phase Estimation Method

The relative position between the camera and the grating within one period of the grid pattern is determined by a DFT-based phase estimation method. Here we discuss the one-dimensional method first [1] and then extend it to the two-dimensional case.

Figure 3(a) shows a periodic one-dimensional signal before and after displacement, which can be represented as a discrete signal:

$$x[n], \quad n = 0, 1, 2, \dots, N-1 \quad (1)$$

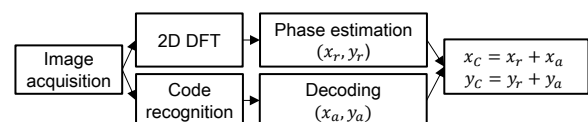


FIGURE 2. Flowchart of the image processing algorithm.

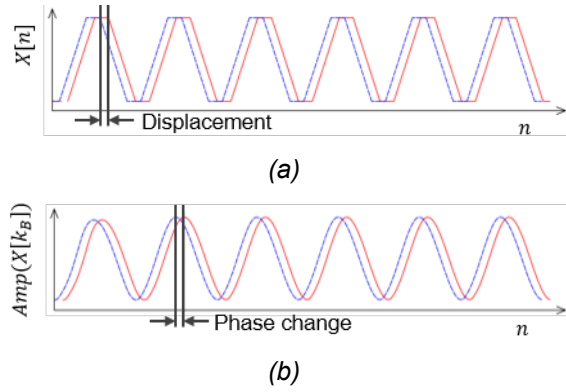


FIGURE 3. (a) One-dimensional periodic signal with displacement. (b) The corresponding fundamental frequency component with phase change.

Applying DFT to $x[n]$, we have

$$X[k] = \sum_{n=0}^{N-1} x[n] e^{-i2\pi \frac{k}{N} n}, \quad k = 0 \text{ to } N-1 \quad (2)$$

where $X[k]$ is the spectrum of $x[n]$.

The aim of the phase estimation method is to determine the phase of the fundamental frequency component of the periodic signal. Let k_B be the index of the fundamental frequency component. Then $X[k_B]$ represents the fundamental frequency component of the signal in the frequency domain. Its phase can be calculated as:

$$\theta_{k_B} = \text{Angle}(X[k_B]) \quad (3)$$

which has a range of $[0, 2\pi)$. This phase value can be converted to a corresponding position value in one period of the grid pattern:

$$x_r = \frac{\theta_{k_B}}{2\pi} T \quad (4)$$

where T is the period of the grid pattern.

When displacement occurs, the signal $x[n]$ is shifted, which causes a phase change in the fundamental frequency component as shown in Figure 3(b). This phase change is proportional to the displacement of the original signal, which can be seen from Eq. (4).

It should be noted that the original signal should be periodic and the period of the grid pattern

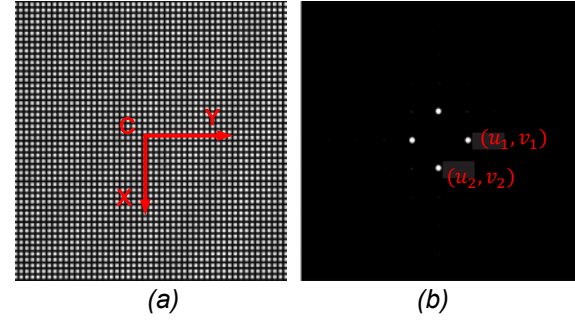


FIGURE 4. (a) Two-dimensional periodic grid pattern. (b) Amplitude spectrum. The white dots are the fundamental frequency components.

should be known. Since the obtained phase is highly sensitive to position change, this phase estimation method allows position measurement to be done at an extremely high resolution.

In the two-dimensional phase estimation method, a two-dimensional periodic grid pattern shown in Figure 4(a) is used. Instead of 1D DFT, 2D DFT is used to determine the phases, θ_x and θ_y , of the fundamental frequency components of the grid pattern along X- and Y-axis. With these phase values known, the position of the image center (x_r, y_r) within one period of the grid pattern can be calculated as:

$$\begin{aligned} x_r &= \frac{\theta_x}{2\pi} T_x \\ y_r &= \frac{\theta_y}{2\pi} T_y \end{aligned} \quad (5)$$

where T_x and T_y are periods of the grid pattern in X- and Y-axis respectively. The fundamental frequency components are located by searching for the first four maximum amplitude values in the spectrum as shown in Figure 4(b). Due to symmetry, two components, (u_1, v_1) and (u_2, v_2) on the X- and Y-axis respectively, are selected and used to determine the phase values via Eq. (3) in both directions.

Absolute Position Coding Method

For absolute position measurement, it is necessary to determine in which period of the grid pattern the image center lies. The idea of the proposed absolute position coding method is to embed position codes into the grid pattern so that the same image can be processed to obtain both (x_r, y_r) and (x_a, y_a) simultaneously. Since phase estimation accuracy depends on the periodicity of the signal, these embedded

position codes should cause as little disturbance to the signal periodicity as possible. This means that the position codes should be distributed sparsely over the entire grid pattern.

In this research, a coding method based on the concept of perfect map is used. An array $(r,s;m,n)$ of a perfect map is an array $(r \times s)$ in which each subarray $(m \times n)$ appears exactly only once. Specifically, a square (256×256) de Bruijn array (a type of binary perfect map) with a (4×4) window property is constructed based on the method proposed by Fan *et al.* [2]. A de Bruijn array with a larger size of $(r, 2^n s; m + 1, n)$ can be constructed from a smaller array $(r,s;m,n)$ and a so-called de Bruijn sequence. Using this method and starting with an array of size (4×16) , we can enlarge the array size step by step using a three-level construction method to build a larger array of size (256×256) .

A decoding method proposed by Shiu [3] can be used to decode a subarray that is selected by a window. The decoding process is an iterative process that follows the three-level construction chain. Decoding at a higher level needs the decoding result of its adjacent lower level. The decoding result is the coordinate of the subarray in the whole code array. Figure 5 shows parts of the $(256, 256; 4, 4)$ array as well as three subarrays or code arrays A, B, and C that are selected by a (4×4) window. The numbers on the left and top of the array are the X coordinates (row numbers) and Y coordinates (column numbers) respectively. The coordinate of a code array is the coordinate of its top-left corner element in the whole (256×256) array. For instance, the decoding result of code array A is coordinate pair (5, 12), which is the coordinate of its top-left element marked by a circle. The decoding process can be expressed as:

$$\text{Decode} \left(\begin{bmatrix} c_{11} & c_{12} & c_{13} & c_{14} \\ c_{21} & c_{22} & c_{23} & c_{24} \\ c_{31} & c_{32} & c_{33} & c_{34} \\ c_{41} & c_{42} & c_{43} & c_{44} \end{bmatrix} \right) = \begin{cases} p : \text{row No. of } c_{11} \text{ in the } (256 \times 256) \text{ array} \\ q : \text{col. No. of } c_{11} \text{ in the } (256 \times 256) \text{ array} \end{cases} \quad (6)$$

Figure 6 shows an image of the grid pattern with the embedded code arrays. Point C is the image center and the small red boxes c_{ij} indicate the elements of a code array. A square represents

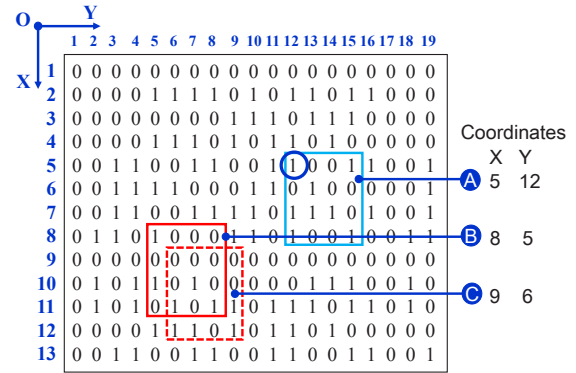


FIGURE 5. Decoding of a subarray to determine its position in the whole array.

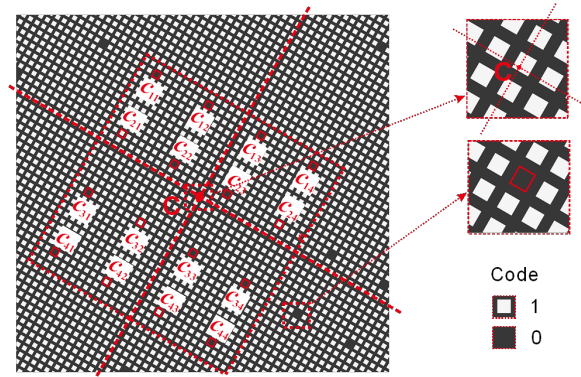


FIGURE 6. Grid pattern with embedded code arrays.

code value 1. If a square is missing, the code value is 0. The distance between two adjacent code elements is called code pitch. For this decoding method to work, at least one complete code array should be visible in the image captured by the camera. In order to reduce the disturbance to pattern periodicity, code elements should be distributed as sparsely as possible. In other words, the code pitch should be as large as possible. In this research, code pitches of $8T_x$ and $7T_y$ in X and Y directions respectively are used. Here different pitches are used to distinguish X and Y directions. This arrangement results in only one complete (4×4) code array visible in the field-of-view of the camera.

Code elements $c_{11} \sim c_{44}$ form a (4×4) code array, which can be decoded to determine (p, q) , the coordinate of c_{11} in the (256×256) array. The position of c_{11} can thus be calculated by the following equations:

$$\begin{aligned} x_{c11} &= 8T_x(p-1) \\ y_{c11} &= 7T_y(q-1) \end{aligned} \quad (7)$$

The position of the image center C can be determined as

$$\begin{aligned} x_a &= x_{c11} + N_x T_x \\ y_a &= y_{c11} + N_y T_y \end{aligned} \quad (8)$$

where N_x and N_y are the numbers of grid periods between the image center C and c_{11} along X and Y directions respectively. The final high-resolution absolute position of the image center C can then be calculated as follows:

$$\begin{aligned} x_C &= x_a + x_r \\ y_C &= y_a + y_r \end{aligned} \quad (9)$$

Rotation Angle Measurement

Rotation angle measurement is based on the absolute linear position measurement method discussed above. Figure 7 shows the concept. Points A and B are centers of two separate regions of the image marked by red rectangles. By applying the above method to these two regions separately, the absolute positions of the points A and B, (x_A, y_A) and (x_B, y_B) , can be determined. From these coordinates, the angle between line AB and Y-axis of the grid pattern θ_Z can be calculated:

$$\theta_Z = \text{atan2}(y_A - y_B, x_A - x_B) \quad (10)$$

If points A and B are in the same row of the camera image, then θ_Z is the rotation angle between the pixel pattern of the camera and the grid pattern of the grating. It should be noted that two separate cameras could also be used to enlarge the distance between points A and B for higher measurement resolution.

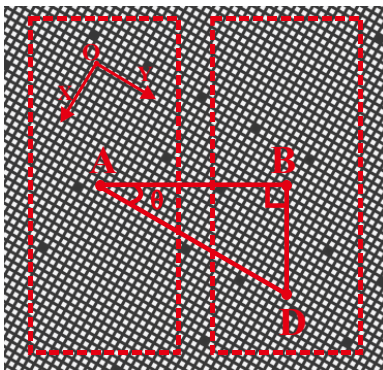


FIGURE 7. Rotation angle measurement.

EXPERIMENTS

Experiments were conducted to evaluate the performance of this absolute planar encoder.

Resolution Evaluation

The first experiment was conducted to evaluate the resolutions of the planar encoder. As shown in Figure 8, the experimental setup includes an industrial B/W camera with 640×480 pixels, a low distortion lens with 1× magnification, a grating with a 100-μm grid pattern and embedded position codes fabricated by a lithography-based micromachining technique, and a piezo-driven $XY\theta_Z$ stage (resolutions: 1 nm in XY and 0.3 μrad in θ_Z directions). The grating is fixed on the stage, which is driven to move step by step along X, Y, and θ_Z directions respectively to test the resolutions of the encoder. At each step, 50 images are taken and processed to obtain the position information. The results are shown in Figure 9. The RMS uncertainty is approximately 1.3 nm in X axis, 1.2 nm in Y axis, and 0.9 μrad in θ_Z axis.

Accuracy Evaluation

The experiment to evaluate the accuracy of rotation angle measurement was conducted using the same experimental setup shown in Figure 8. The stage was controlled to move at a step of 50 μrad and in a range of -500 μrad to +500 μrad. The angular positioning accuracy of the stage in this range is 0.3 μrad. Figure 10 shows the angle measurement error or the difference between the angle measured by the planar encoder and the angular position of the stage. We can see that the error is less than 1.3 μrad over a measurement range of 1000 μrad. Also, the error increases almost linearly with the stage rotation angle, which could be caused by the misalignment of the grating plane and the stage plane.

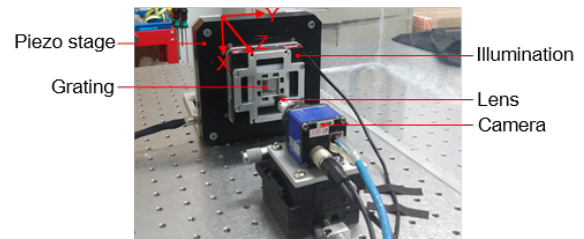
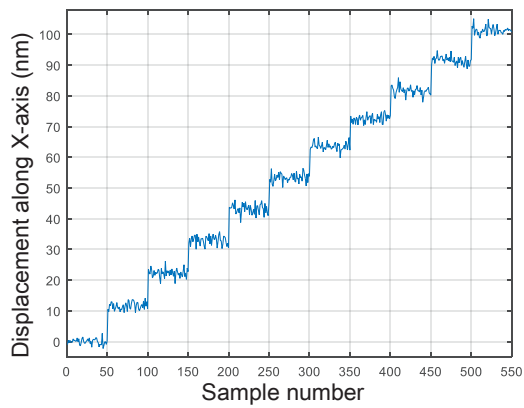
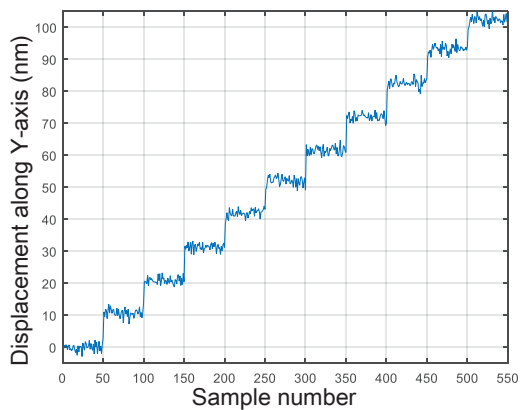


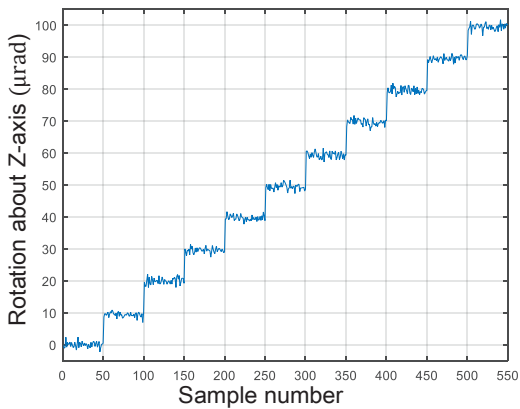
FIGURE 8. Experimental setup for resolution evaluation.



(a)



(b)



(c)

FIGURE 9. Results of the experiment for resolution evaluation: (a) X axis; (b) Y axis; (c) θ_z axis.

Another experiment was conducted to evaluate the accuracy of linear position measurement of the planar encoder. Figure 11 shows the experimental setup. The grating is fixed on a linear stage with a travel range of 100 mm and a

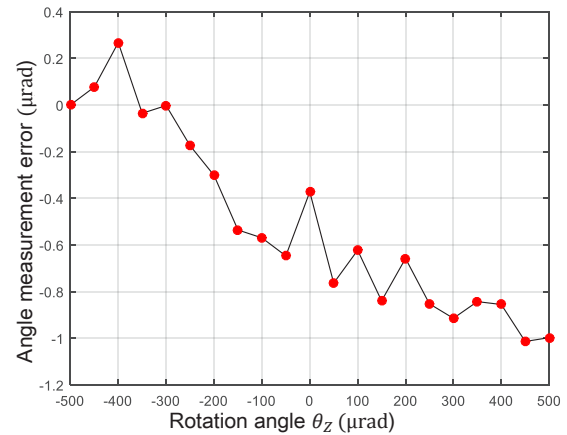


FIGURE 10. Result of the experiment for evaluation of angle measurement accuracy.

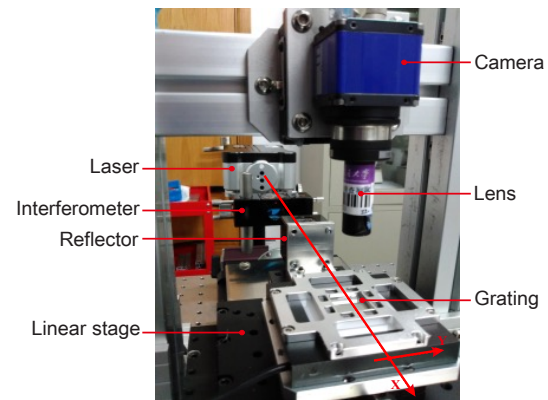


FIGURE 11. Experimental setup for evaluation of linear position measurement accuracy.

resolution of $2.5 \mu\text{m}$. The displacement of the grating or the linear stage is measured by the planar encoder and a laser interferometer simultaneously for comparison.

Figure 12 shows the results. The horizontal axis represents the nominal position of the linear stage from the controller of the stage. The vertical axis on the left hand side represents the linear positioning error of the stage, or the difference between the measured position by the planar encoder or the laser interferometer and the nominal position. The vertical axis on the right hand side shows the residual or the difference between the measured positions by the planar encoder and the laser interferometer. The planar encoder was calibrated by the interferometer beforehand. The results show that the residual of the measurements is within $\pm 0.2 \mu\text{m}$.

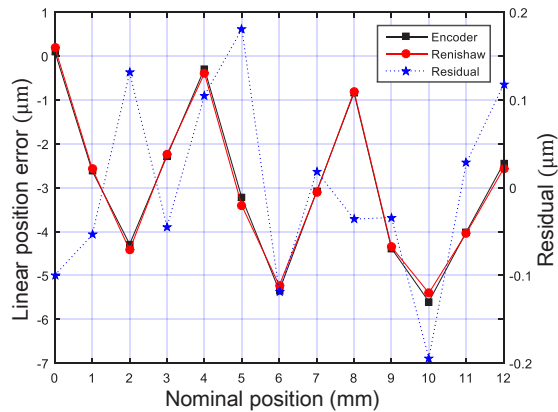


FIGURE 12. Results of linear position measurement of a linear stage.

CONCLUSION

This paper presents the design of a novel vision-based planar encoder for high precision in-plane absolute position measurement. A phase estimation method and an absolute position coding method were proposed for simultaneous and high-resolution measurement of absolute $XY\theta_z$ positions. Experimental results showed that this encoder could achieve nanometer order resolution for linear position measurement and sub-micro-radian level resolution for angular position measurement. Experimental results also demonstrated a measurement accuracy of $\pm 0.2 \mu\text{m}$ over 12 mm for linear position measurement and $1.3 \mu\text{rad}$ over 1000 μrad for rotation angle measurement.

ACKNOWLEDGEMENT

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CHARACTERISING 3D OPTICAL SCANNER MEASUREMENT PERFORMANCE FOR PRECISION ENGINEERING

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ABSTRACT

This report summarises test results from the National Physical Laboratory's (NPL, UK) recently launched 3D optical scanner dimensional characterisation facility. This facility comprises a purpose-built environmentally controlled laboratory with test artefacts, test procedures and equipment to assess 3D optical scanner performance. The reported test results include an indication of 3D optical scanner resolution and changes in measured dimensions due to: instrument temperature, artefact illumination, scanner orientation, artefact surface finish and artefact position within a measurement volume. NPL's 3D optical scanner facility has been developed to assist the rapid take-up of 3D optical scanners on production lines of metre-scale precision components manufactured by the aerospace, automotive and power generation industries, by promoting best measurement practice and to support measurement traceability.

INTRODUCTION

3D optical scanners are rapidly replacing Cartesian CMMs on the workshop floors of the aerospace, automotive and power generation industries. While not offering equivalent levels of accuracy, 3D optical scanners, such as fringe projectors and laser scanning articulating arms, have benefits over Cartesian CMMs that include portability, fast measurement capability, point cloud data capture and relative ease when measuring freeform surfaces [1].

Cartesian CMMs are well-established [2] within precision engineering, typically capable of micrometre level and in some cases, sub-micrometre level measurement uncertainties. ISO 10360 describes methods for accepting and verifying the performance of Cartesian CMMs [3] and provides end-users with a means of comparing the performance of different instruments, aiding selection of an appropriate instrument for a particular task. Techniques have been established to determine

Cartesian CMM measurement uncertainties using a substitution method based on calibrated artefacts and virtual CMM techniques [4, 5] to provide traceability to the SI.

In contrast, portable 3D optical scanners are a relatively new technology and ISO standards to provide verification and acceptance tests are yet to be published. 3D optical scanner measurement error sources are still being understood and, therefore, techniques to provide measurement traceability are still being developed [6]. The German guideline for optical 3D measuring systems, VDI/VDE 2634 [7], attempts to provide a means of comparing performance between different systems based around optically cooperative artefacts, but does not address freeform surfaces, surface finish nor instrument measurement performance while operating in unfavourable environments. This lack of information can hamper an organisation trying to acquire a suitable 3D optical scanner for their application.

NPL'S CHARACTERISATION FACILITY

To support industry's increasing use of and improve measurement confidence in the technology, NPL has recently launched a 3D optical scanner characterisation facility [8]. It comprises a purpose-built environmentally controlled laboratory (approximate dimensions: 3 m × 5 m × 2.5 m), test artefacts, test procedures and equipment to examine the performance of 3D optical scanners. With the aim of simulating typical environments that 3D optical scanners are used, such as the workshop, where ambient lighting and temperature may be adjusted. The test artefacts have been developed to fit within the ~ (400 × 400 × 300) mm measurement volume that is common to fringe projectors and the test procedures have been designed to verify instrument manufacturer's claimed measurement accuracies, which are typically < 10 µm for fringe projectors and < 100 µm for laser scanning articulating arms.

RESOLUTION INDICATION

NPL has developed an artefact to aid understanding of 3D optical scanner resolution. The artefact, shown in figure 1 (left), comprises an aluminium plate with dimensions (170 × 170 × 45) mm and a circularly-symmetric pattern cut into its top surface. The circular pattern is based upon the zero-order Bessel function of the first kind and provides a series of ripples with an exponential decay in amplitude and frequency with increasing distance from the plate's centre.

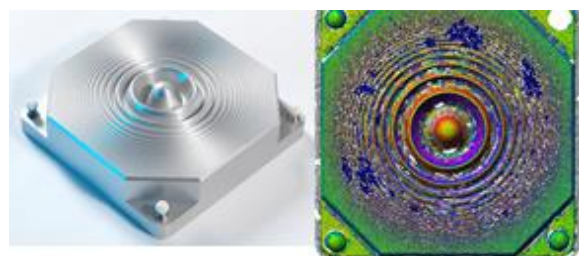


FIGURE 1: (Left) NPL's resolution indication artefact. (Right) Example comparison of 3D scanner measurements against the corrected CAD model, plotted using Polyworks 2015 [9].

Being a complex shape to manufacture, the artefact has been measured at NPL using high precision a Cartesian CMM and a profilometer and the CAD model adjusted accordingly. An indication of the minimum, micrometre scale, feature size that a 3D optical scanner can measure is achieved by comparing scanner measurements of the artefact against the CAD model and analysing the measured circular pattern with increasing distance from the artefact's centre. Figure 1 (right) shows a colour map that gives an example of such a comparison. In the colour map, the colour green indicates that the CAD and measurement compare well; red and blue indicate that a surface has been measured, respectively, above and below the surface of the CAD model.

ORIENTATION EFFECTS

Optimal 3D optical scanner measurement may often require movement of the scanner through a wide range of orientation angles. Measurement and calibration routines for automated measurement systems, which are becoming increasingly common in the automotive industry and usually comprise fringe projectors attached to robotic arms [10-12], may identify anomalies caused by orientation that a human operator may not.

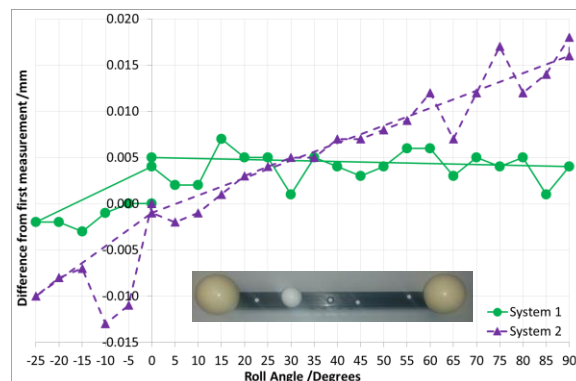


FIGURE 2: The effect of roll angle on dimensional measurements. Inset: the orientation artefact (≈ 270 mm in size).

To aid understanding of how measurements can be affected by scanner orientation, NPL uses a precision mount that allows a ballbar artefact to be positioned at the centre of the scanner's measurement volume and follow the scanner through a range of orientation angles. The ballbar artefact (figure 2) comprises three spheres. The test involves measuring the ≈ 270 mm separation between the centres of the 47.5 mm diameter spheres at each end of the artefact as the pitch or roll angle of the scanner is varied, from one extreme (identified as -25° from horizontal roll or vertical pitch) to the other extreme (90°) in 5° increments. To simulate typical usage, the scanner is adjusted according to the manufacturer's instructions before the measurements commence. Automated analysis of the captured point cloud representing the artefact at each angle is performed using a Polyworks 2015 [9] macro to identify changes in the measured sphere separation, with orientation angle.

Figure 2 presents the effects of roll angle on two different commercial white light fringe projection systems; in the interests of impartiality, the models have not been identified. System 1 appears unaffected by roll angle, the measured differences in sphere distances being well within the expected performance of the instrument. System 2, however, appears much more sensitive to roll angle, showing a difference of ≈ 25 μm between the two extremes in angle and is outside the expected performance of the instrument. Possible causes for this change may include flexing of the scanner's frame or slight movement within the scanner's internal optics.

ILLUMINATION EFFECTS

Ambient lighting has been shown to affect 3D optical scanners [13, 14]. Therefore, it is reasonable to expect the ambient lighting in the industrial environments, where 3D optical scanners are increasingly being used, to affect measurements.

The ambient lighting within NPL's 3D optical scanner characterisation facility is adjustable to simulate typical usage environments, allowing assessment of illumination effects on scanner measurements. Illumination conditions include: no illumination; typical workplace lighting conditions (≈ 650 lx); high pressure sodium lamp illumination to simulate certain workshop floor environments; variable strobe lighting to simulate flashes from welding processes and gradually changing lighting.

Assessment of the effects of illumination on 3D optical scanner measurements is performed using an artefact that comprises fifteen 50.8 mm diameter coloured (red, blue and yellow) spheres in a triangular arrangement, such that each sphere is in contact with its neighbours (figure 3).

Measurements are performed by first optimising the scanner according to the manufacturer's instructions. The scanner is aligned centrally with the sphere labelled "5" (figure 3) and oriented at an angle perpendicular to all of the spheres. Scans of the artefact are then performed under different lighting conditions, using optimal 3D optical scanner measurement parameters. A calibrated Gigahertz-Optik spectrophotometer is used to record the illumination in the plane of the artefact.

Figure 3 presents the differences in the measured diameters of blue spheres under typical workplace lighting when compared with measurements under no additional lighting (the artefact was illuminated by the 3D optical scanner only). The measurements were performed using a white light fringe projector with an approximate $(400 \times 400 \times 300)$ mm measurement volume. The diameters of the blue spheres are measured to be larger, by up to $260 \mu\text{m}$, when illuminated by typical workplace lighting compared with 3D optical scanner only illumination (no additional lighting). Possible causes include filters within the 3D optical scanner, biasing its measurement towards certain colours, shadows and the interaction of

light in the gaps between the spheres affecting measurements. A more in depth analysis is to be reported in a future paper on the facility.

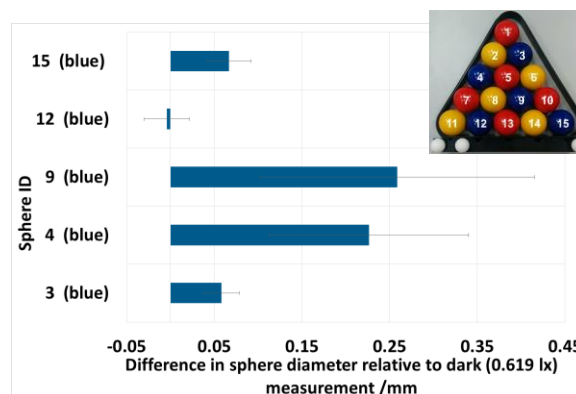


Figure 3: Differences in the measured blue sphere diameters under typical workplace lighting conditions compared with no illumination. Error bars denote measurement repeatability. Inset: the NPL illumination artefact.

VOLUMETRIC ERROR SOURCES

Based on earlier tetrahedron designs [15], NPL has developed a tetrahedron (figure 4) that aims to combine quick, VDI/VDE 2634 style performance verification with assessment of chromatic sensitivity using coloured spheres [16].

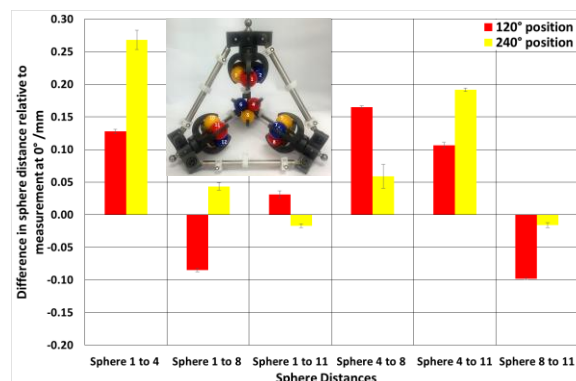


FIGURE 4: Differences in the measured sphere centre distances for the tetrahedron's red spheres. Error bars denote measurement repeatability. Inset: the NPL tetrahedral artefact.

Three sets of precision steel connecting rods allow the tetrahedron to be constructed to fit within typical industrial measurement volumes ranging from $(300 \times 300 \times 300)$ mm up to $(1000 \times 1000 \times 1000)$ mm. Specially designed claws at each of the tetrahedron's corners hold three coloured 50.8 mm diameter spheres, such that they are all in contact.

Measurement of the NPL tetrahedral artefact is based on an established third-party procedure [17]. When assessing a scanner, the tetrahedron is first built to nominally fill the scanner's measurement volume and then placed on an automated rotation stage. The scanner is optimised according to the manufacturer's instructions, oriented at a vertical angle of approximately 15° and placed in front of the tetrahedral artefact such that it is able to view and measure data from all of the spheres in one measurement; this is recorded as the 0° position on the rotation stage. Test measurements are performed and the scanner settings are adjusted to ensure measurements with the best data quality and the maximum data quantity. Once the scanner has been optimised, the tetrahedral artefact is measured at 0° , 120° and 240° positions by turning the rotation stage.

Figure 4 presents results from measurements of the centre distances of the tetrahedron's red spheres when measured using a white light fringe projector with an approximate ($400 \times 400 \times 300$) mm measurement volume. Respective differences in the distances between the red sphere centres when measured at 0° increased by as much as $\approx 150 \mu\text{m}$ and $\approx 250 \mu\text{m}$ when in the 120° and 240° positions, suggesting that the XYZ positions of the measured sphere centres vary significantly between the tetrahedron's rotation positions. A future paper will report more detailed analysis of the NPL tetrahedral measurements.

Variations in sphere centre distances at different positions within the measurement volume are to be expected. Possible causes for these variations include aberrations in the scanner's optics, partial occlusion of the spheres by neighbouring spheres, and colour bias caused by filters within the instrument.

SURFACE COLOUR, MATERIAL AND FINISH

The optical properties of an object's surface are known to affect 3D optical scanner measurements [18]. To quantify a 3D optical scanner's ability to measure different surface finishes (material, reflectance, roughness and colour), NPL has developed the 3D material coupon plate, a multi-faceted test artefact [19].

The NPL 3D material coupon plate comprises three rows and five columns of (60×60) mm square coupons; a 170° angle separates each column rather than forming a continuous plane.

Five highly Lambertian, sintered PTFE (Spectralon®) reference coupons are placed along the plate's central row, while the sample coupons are placed in the rows above and below. Sample materials investigated include: carbon-fibre-reinforced polymer, steel foil, various aluminium finishes, colour (using RAL colour standards [20]) and surface roughness (using different grades of sandpaper).

A scanner under investigation is first optimised according to the manufacturer's instructions. It is then aligned with the centre of the middle reference coupon on the 3D material coupon plate, such that it is normal to the coupon's surface. Measurements are performed using an alignment jig to allow each of the artefact's coupons to be measured normal to, at $\pm 10^\circ$ and $\pm 20^\circ$ relative to the scanner, using optimal measurement parameters.

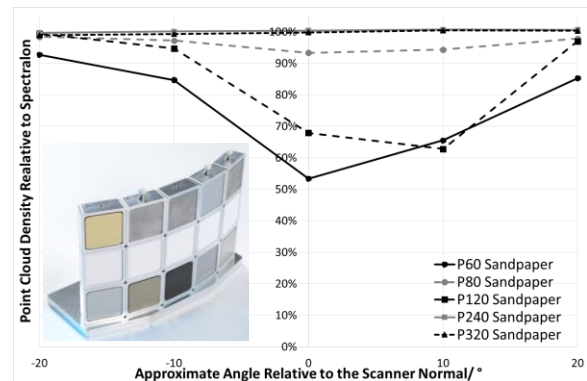


FIGURE 5: Normalised point cloud densities for samples of different surface roughness. Inset: the NPL 3D material coupon plate.

The captured point clouds are subsequently analysed using Polyworks 2015 [9] to measure the number of points within a 30 mm by 30 mm region within the centre of each coupon. The number of points within each region is normalised against an identical region within the respective Spectralon reference coupon for the column. Thus, data quantity is measured, determining a scanner's ability to measure a particular material or surface finish at a particular angle.

Figure 5 presents results from measurements using a white light fringe projector of coupons containing different sandpapers, with ISO grit designations [21], to measure surface roughness effects. The results may suggest a trend of increasing point cloud density with decreasing surface roughness.

TEMPERATURE EFFECTS

Temperature is a significant source of uncertainty in dimensional measurements. Therefore, NPL's 3D optical scanner characterisation facility has been designed to simulate typical usage conditions [22] by controlling the air temperature across a range from 16 °C to 24 °C, to allow assessment of temperature effects.

Temperature effects on 3D optical scanner measurements are assessed using a ballbar artefact (inset in figure 6) that comprises two 50.8 mm diameter spheres with a ≈ 300 mm separation between their centres attached to a Zerodur bar; the Zerodur has a thermal expansion coefficient of 0.01 ppm/°C.

Tests are performed by setting the facility air temperature to a particular set point temperature, either 16 °C or 24 °C, and allowing the artefact and the scanner under test to reach a stable temperature within the facility. Once stable, the scanner is optimised according to the manufacturer's instructions and set to repeatedly measure the artefact, at 15 minute intervals. The set point is then adjusted to the other temperature extreme (24 °C or 16 °C) and the measurements are continued until the scanner has stabilised to the new temperature. Once the scanner is stable at the new temperature, the process is repeated in the opposite temperature direction. Polyworks 2015 [9] is used to measure the distance between the sphere centres in the captured point clouds.

Figure 6 presents the measurement results from two carbon-fibre framed white light fringe projecting 3D optical scanners, for reasons of impartiality referred to as systems A and B. System A measured a 60 μm increase and then a 50 μm decrease in the sphere centres separation after increasing and then decreasing the facility's set point temperature. Measurements on system B resulted in a 50 μm increase and then a 30 μm decrease in the sphere centres separation after increasing and then decreasing the facility's set point temperature.

From calculation, the length of the Zerodur bar is expected to change by 31 nm across this temperature range. Therefore, the results strongly suggest that the instruments are the cause of these dimensional changes. Differences between the measurements for the

upward and downward temperatures changes may have been caused by hysteresis in each system's frame, their optical component dimensions or positions, or projector and camera angles.

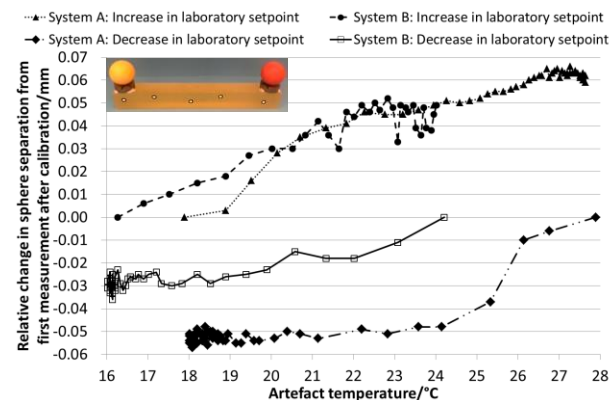


FIGURE 6: The effect of temperature on dimensional measurements from systems A and B. Inset: the zerodur ballbar artefact.

CONCLUSIONS

NPL has developed a 3D optical scanner dimensional characterisation facility to provide global industries with greater confidence in 3D optical scanners, which are rapidly replacing traditional Cartesian CMMs as they offer point cloud data capture, portability and speed. The NPL characterisation facility contains test artefacts, test procedures and equipment to examine the performance of 3D optical surface-form scanners. This paper summarises the facility's test procedures and gives an overview of results from instruments so far tested within the facility. The presented results show how NPL assesses scanner resolution and confirm that temperature, illumination, instrument orientation, object surface finish and position within the measurement volume can each affect 3D optical surface measurements.

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DISPLACEMENT-BASED MEASUREMENT OF STATIC AND DYNAMIC COEFFICIENTS OF FRICTION

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INTRODUCTION

Experimental methods are commonly used to identify friction behavior, particularly for manufacturing processes [1-22]. The majority of these friction measurements are made on tribometers, which apply a normal force to a pin that is pressed against a rotating or translating surface. The normal and friction forces are measured and the friction coefficient is calculated. However, misalignment between the force transducer axes and the motion and the applied direction of the normal force are critical factors in establishing the accuracy of the derived coefficients of friction [23].

Current methods used to measure friction behavior and parameterize complex friction models can suffer from high uncertainty, especially for low friction interfaces. Relative uncertainties on the order of two to four parts in 10^2 of the measured value are typical for low friction interfaces [23]. In contrast, displacement can be measured by laser interferometry with relative uncertainties on the order of a few parts in 10^7 , four to five orders of magnitude better. The research described in this paper will evaluate a fundamentally new method for measuring friction and parameterizing friction models with lower uncertainty than conventional methods. The innovation is the use of high-accuracy displacement and velocity measurements during dynamic motions to characterize the energy dissipation.

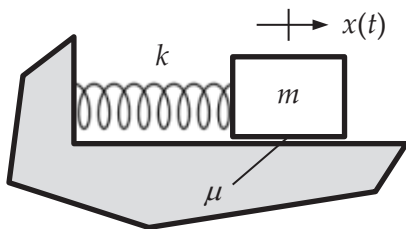


Figure 1. Spring-mass oscillator with Coulomb friction.

MEASUREMENT THEORY

To find the coefficient of friction between two materials, a normal force is applied between them and relative motion is provided. As shown in Fig. 1, the moving mass, m , is given an initial displacement

and decaying oscillating motion occurs due to the linear spring, k , and Coulomb (frictional) damping, μ . Given an initial displacement of the moving mass, a final displacement is obtained. In the proposed instrument, displacement is recorded using displacement measuring interferometry.

SIMULATION RESULTS

Sliding, or Coulomb, friction can be incorporated into the equation of motion describing the time-dependent displacement, $x(t)$, of the Fig. 1 spring-mass system as shown in Eq. 1. In this equation, F_f is the friction force (i.e., the product of the friction coefficient, μ , and the normal force, $N = mg$).

$$\begin{aligned} m\ddot{x} + kx + F_f &= 0, \dot{x} > 0 \\ m\ddot{x} + kx &= 0, \dot{x} = 0 \\ m\ddot{x} + kx - F_f &= 0, \dot{x} < 0 \end{aligned} \quad (1)$$

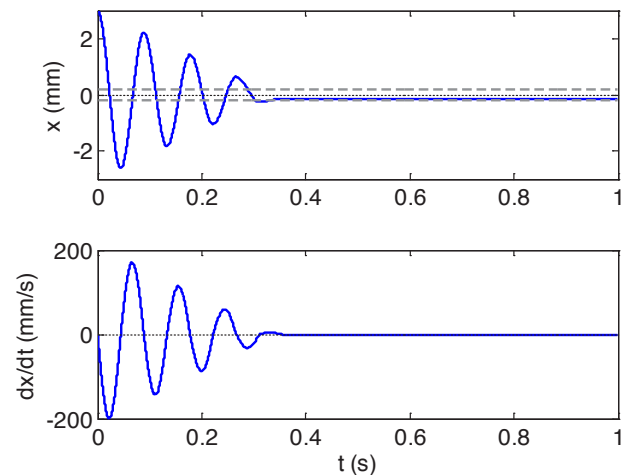


Figure 2: Free vibration result for $x_0 = 3$ mm.

Because the friction force always opposes the velocity direction, it is discontinuous. This yields the nonlinear second order, homogeneous differential equation shown in Eq. 1. For the displacement-based approach to be used in this research, the desired information is $x(t)$ (as well as its time derivative, or velocity). The relationship between the initial mass

displacement, $x(0) = x_0$, and the corresponding motion will be established. Specifically, due to the friction force, the final displacement, x_f , depends on x_0 .

Consider the case where $\mu = 0.1$, $m = 1$ kg, and $k = 5 \times 10^3$ N/m for the model in Fig. 1. The free vibration response for $x_0 = 3$ mm (zero initial velocity) is displayed in Fig. 2, where $x_f = -0.139$ mm. It is observed that when the velocity reaches zero (lower panel near 0.1 s), if the current displacement is between $x_{lim} = \frac{F_f}{k} = \frac{\mu N}{k}$ and $-x_{lim}$, the motion stops. This limiting displacement (marked by the horizontal dashed lines in the top panel of Fig. 2) is the x value where the spring force is equal to the friction force. For a new initial displacement of $x_0 = 1$ mm, the final mass position is $x_f = 0.176$ mm; see Fig. 3.

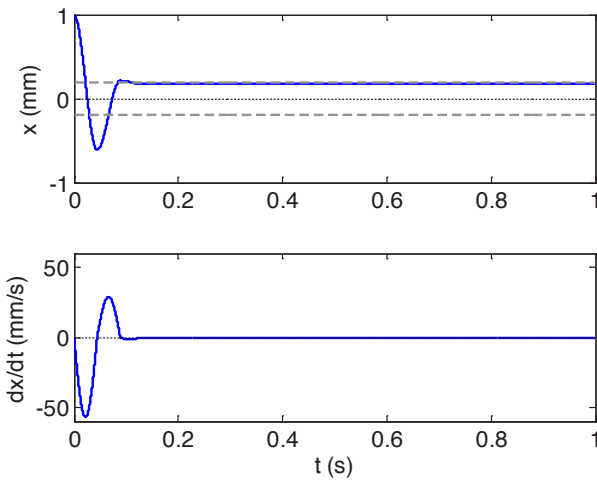


Figure 3: Free vibration result for $x_0 = 1$ mm.

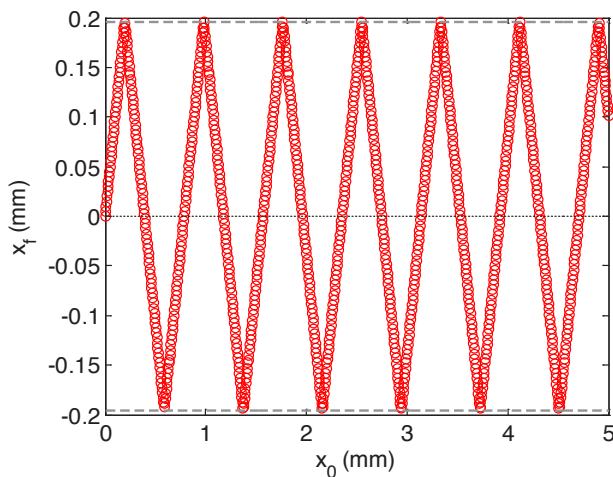


Figure 4: Initial vs. final displacement (Coulomb friction).

If the final displacement is plotted versus the initial displacement for this system, a periodic structure is revealed. This result is presented in Fig. 4. The period of the triangular waveform is $\frac{4F_f}{k} = \frac{4\mu N}{k}$ and its amplitude is x_{lim} .

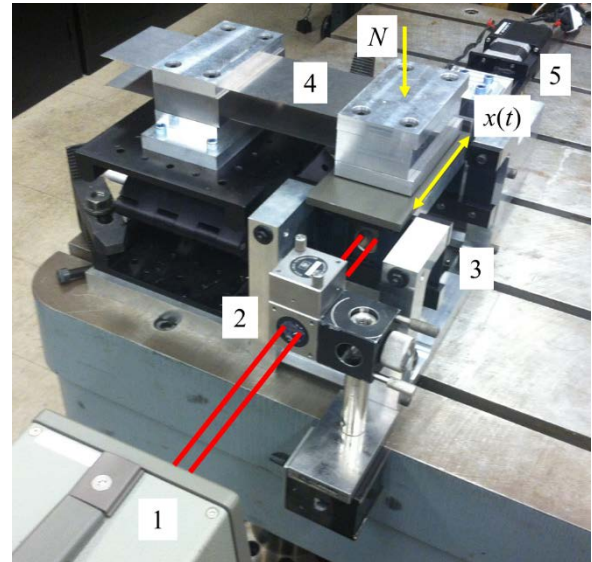
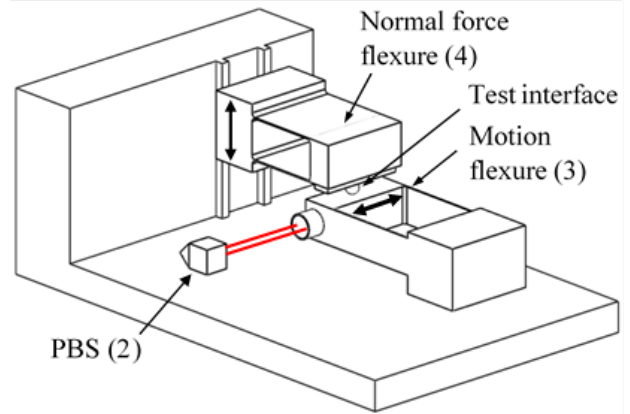


Figure 5: Schematic (top) and photograph (bottom) of prototype setup including: (1) DMI; (2) PBS; (3) motion flexure; (4) normal force flexure; (5) motor/stage/electro-magnet for initial displacement.

INITIAL EXPERIMENTAL RESULTS

To provide preliminary test data, a prototype setup was designed and constructed. The design concept used flexures for both the normal force application and the free vibration motion direction. Flexures were chosen to constrain the motion (ideally) to a single direction with low stiffness and a selectable natural frequency [24]. The setup is displayed in Fig. 5. The primary components were:

- displacement measuring interferometer (DMI) to measure the motion flexure displacement, x_0 and

$x(t)$; the DMI includes the laser head, polarizing beam splitter (PBS), and retroreflector (attached to the underside of the flexure in the Fig. 5 photograph)

- motion flexure – this provides the relative motion between the sample and counterface
- normal force flexure – this flexure was used to provide the prescribed normal force, N , between the contact surfaces (it is very stiff in the x direction, but flexible in the vertical direction); masses were stacked on the flexure to change the normal force
- motor/stage/electro-magnet for initial displacement – the electromagnet was energized to pull the flexure away from its equilibrium position to the prescribed initial displacement and then turned off to release the flexure and initiate free vibration.

The flexure dynamics were measured by impact testing, where an instrumented hammer is used to excite the structure and a linear transducer is used to record the corresponding motion [25]. This time domain data is converted to the frequency domain and the displacement-to-force ratio is calculated. This frequency response function can then be fit using modal analysis techniques to identify the natural frequency, stiffness, and damping ratio. In this case, the flexure had a dominant natural frequency of 12.6 Hz with a stiffness of 2574 N/m and a low viscous damping ratio of 0.0027 (0.27%).

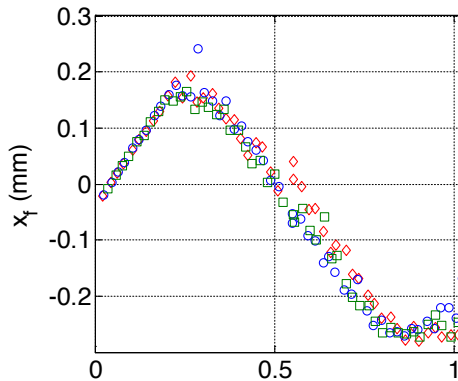


Figure 6: Initial vs. final displacement for UHMWP on anodized aluminum (three repeated trials).

In a first series of tests, the normal force between a flat pin (ultra-high molecular weight polyethylene, UHMWP, with a small percentage of polytetrafluoroethylene, PTFE, added) and an anodized aluminum counterface was set to be 9.66 N, initial displacements of 0.025 mm to 1.0 mm were imposed, and the final displacement was identified from the free oscillation response measured by the DMI. The

results of three repeated trials are presented in Fig. 6, where the horizontal axis is the initial displacement for the motion flexure mass/counterface and the vertical axis is its final displacement after oscillation. The triangular waveform shown in Fig. 4 was observed, so these promising initial results warrant further investigation.

In a second series of tests, rolling friction was examined by inserting a 4.76 mm diameter steel sphere between the UHMWP and anodized aluminum surfaces. Figure 7 shows a comparison of the free vibration for a 2 mm initial displacement with contact (sphere inserted with a normal force of 10.89 N) and no contact (flexure motion only). This increased energy dissipation due to friction is observed and demonstrates that this setup (or a modification) can be used for low friction interfaces.

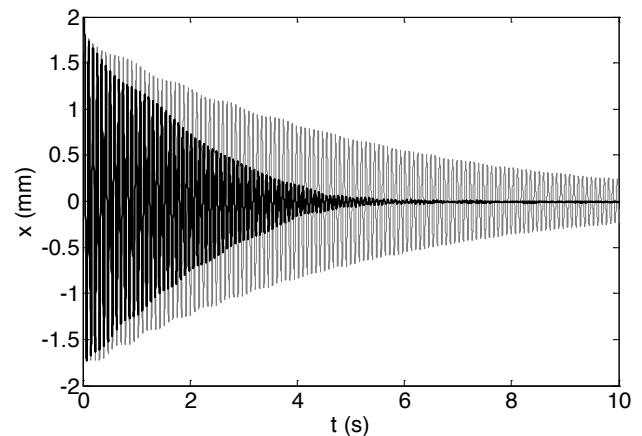


Figure 7: Free vibration for (gray) flexure only; and (black) rolling steel sphere between UHMWP and aluminum.

NEW DESIGN

A number of design improvements are planned for the new friction measuring machine:

- increase the motion flexure leaf spring length to decrease parasitic motions and reduce structural damping
- place the intersection of the measurement, pin/counterface contact, and force lines at the center of the flexure body to eliminate moments and Abbé error
- replace the normal force flexure with an air bearing
- use a large angle plate as the motion flexure mount.

A model of the new design is displayed in Fig. 8, where the angle plate supports the (vertical) force application to the pin using an air bearing and the base of the four flexure leaf springs that provide

(horizontal) linear motion of the motion platform (counterface). In operation, the motor will extend the stage that supports the electro-magnet (through a decoupling flexure) toward the motion platform. The electro-magnet will be energized to capture the motion platform. The motor will then retract the stage to provide the initial displacement of the motion platform (through leaf spring deflection). By de-energizing the electro-magnet, the motion platform will be released and will oscillate in the horizontal plane. The DMI will be used to measure the initial, time-dependent, and final platform displacements (the moving retroreflector will be attached to the motion platform; the DMI measurement axis will pass through the pin-counterface contact point to eliminate Abbé effects). The normal force will be varied by changing the deadweight mass on top of the cylinder that passes vertically through the (fixed) air bearing. The pin will be mounted on the bottom of this cylinder. The (vertical) normal force will press the pin against the counterface, which will be attached to the motion platform.

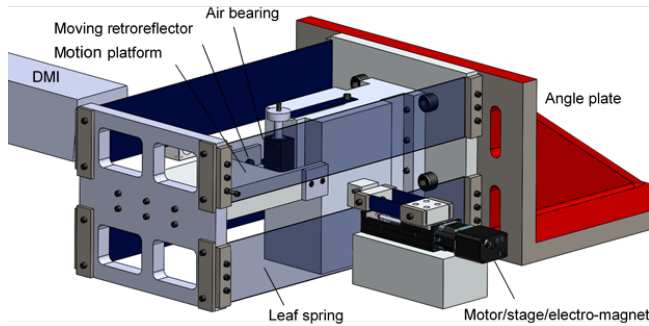


Figure 8: New friction measuring machine design.

DATA EVALUATION

A primary objective of this research is to enable friction model parameterization with increased accuracy. Therefore, the data evaluation will include the selected friction model, the measured displacement and velocity, and the friction measuring machine structural dynamics. The friction model will describe the friction force acting between the pin and counterface and will appear in the second order differential equation that describes the flexure motion:

$$m\ddot{x} + c\dot{x} + kx + F_f \text{sgn}(\dot{x}) = 0, \quad (2)$$

where c is the viscous damping coefficient (due to the structural energy dissipation) and the signum function captures the velocity sign dependence of the friction force. To assign values to the friction model coefficients, the measured displacements (initial, time dependent, and final) will be compared to the solution of Eq. 2. This comparison can take several forms.

- An over-constrained optimization approach can be applied to identify the best-fit friction model parameters by minimizing the difference between the measured displacement and $x(t)$ and the Eq. 2 solution. The friction model can be identified from a single initial displacement by this technique. Multiple initial displacements will be used to verify consistent results.
- The global initial vs. final displacement behavior shown in Figs. 4 and 6 can be used to parameterize the friction model. In one scenario, multiple experiments will be used to determine {initial displacement, final displacement} pairs and the optimization problem will minimize the difference between these pairs and the corresponding Eq. 2 solution pairs. In other words, the friction parameters will be selected to achieve a best-fit between the simulated plot (Fig. 4) and its experimental counterpart (Fig. 6). In a second approach, the period and amplitude of the experimental initial vs. final displacement graph will be used to define the friction model parameters.
- The DMI provides high accuracy measurement of the instantaneous position and velocity of the moving counterface. Since the total energy in the system is the sum of the kinetic energy in the moving mass ($\frac{1}{2}mv^2$) and the potential energy stored in the flexure springs ($\frac{1}{2}kx^2$), it will be possible to track the instantaneous rate of energy dissipation in the system. When the viscous (and windage) losses are subtracted using data from free vibration of the system with no frictional contact, it will be possible to map the instantaneous rate of frictional energy dissipation over the entire experiment. This will enable an accurate quantification of variations in the frictional force over the full range of sliding velocities, including behavior near zero velocity. Simulated energy dissipation results are presented in Fig. 9 for $\mu = 0.01$, $m = 1$ kg, $k = 5 \times 10^3$ N/m, a 1% viscous damping ratio, and $x_0 = 3$ mm. As seen in the top panel, for no damping or friction, the energy is constant. The energy loss is exponential for viscous damping only and parabolic for friction only. It is a combination when both are present. In the bottom panel, it is observed that the energy dissipation rate increases from none to friction only to damping only to both and varies with velocity. The friction energy rate, FER in the figure, will be identified by subtracting the damping energy rate (measured with no friction contact) from the combined

energy rate. The friction force will be determined by dividing FER by the sliding velocity.

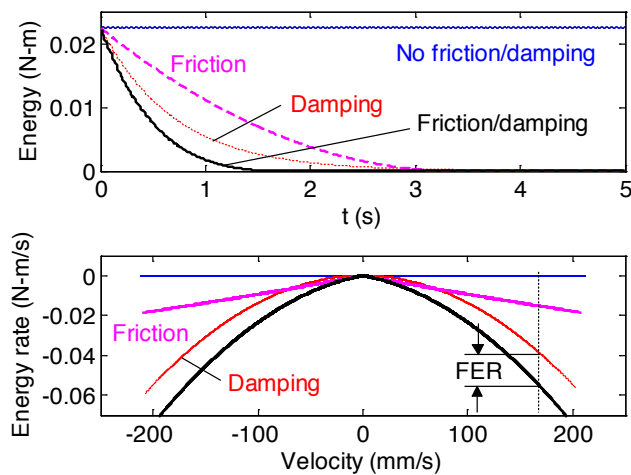


Figure 9: (Top) Energy loss vs. time; (bottom) energy loss per unit time vs. sliding velocity.

CONCLUSIONS

The goal of this research is to increase friction measurement accuracy with a particular emphasis on low friction applications. This will enable: improved modeling, prediction, and control of manufacturing processes; creation and evaluation of new manufacturing coatings and lubricants; and design of new low friction elements, including bearings of all types, to be used in manufacturing machines. To realize this goal, a novel displacement-based, dynamic measurement approach is being tested. This offers a new Lagrangian strategy with a focus on energy dissipation, as opposed to the traditional force-based Newtonian method for current tribometers.

This paper presented both simulated and experimental results to demonstrate the efficacy of the new displacement-based measurement strategy. A new friction measuring machine design was presented. Once it is constructed, new measurement results will be presented and the measurement uncertainty will be evaluated.

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A GAUSSIAN PROCESS BASED DATA MODELLING AND FUSION METHOD FOR MULTISENSOR COORDINATE MEASURING MACHINES

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INTRODUCTION

Multisensor measurement technology is an emerging technology which makes use of combinations of two or more different types of sensor probes so as to further enhance the measurement capability of the traditional single sensor coordinate measurement machines (CMMs). The sensors can complement each other's limitations and improve the measurement accuracy. Nowadays, the applications of multisensor CMMs are becoming more and more widespread and many CMM manufacturers are developing multisensor CMMs in their advanced production lines. For instances, ZEISS O-INSPECT [1] equips with a contact sensor, imaging sensor and white light distance sensor, which is able to provide a fast inspection by the image sensor and high accuracy 3D measurement results by the contact sensor and white light distance sensor. Werth VideoCheck [2], is designed to equip with many kinds of sensors such as trigger probe, fiber probe and video sensor which provides the measurement ability of small features with the help of the small-diameter fiber probe in the scale down to 20 μm , as well as a quick checking with the fast trigger probe and image sensor. Hexagon Optiv Classic [3] provides a vision sensor and a touch trigger probe, while Nikon [4] enhances the true 3D multi-sensor measurement by combining vision sensor, laser auto-focus sensor, tactile sensor and rotary indexer. The measurement range, resolution and flexibility are largely enhanced by the complementarity of the different characteristics of various sensors.

The combination of different types of sensors extends the measurement ability such as accuracy and measurement range of the CMMs. However, most of the multisensor CMMs are lack of an optimal strategy to perform multisensor measurement and fusion of data from different sensors. Some of the studies for multisensor CMM focused on complementary measurement for special geometrical features. Nashman et al. [5] used a camera sensor to locate and measure the feature such as object edges, corners and centroids while the touch sensor was used to measure other part of the object. The touch sensor was highly

accurate with little noise. However, it could not measure sharp features such as edges and corners. Combining these two sensors enable the capability to gather high bandwidth global information and to obtain high accurate measurement information. Zexiao et al. [6] used a multi-probe system which consists of a structure light sensor and a trigger probe to measure multiple features including edges. However, the edges were not directly measured while they were generated by fitting the surfaces using the measured points on the relatively smooth surfaces instead.

This paper presents a Gaussian process based data modelling and data fusion (GP-DMF) method which first estimates the mean surfaces and uncertainties of the datasets obtained from different sensors and combines the two measurement data into a single one with associated uncertainty. A series of simulation and measurement experiments have been conducted to verify the technical feasibility of the method. The results show that the fused data with a lower uncertainty are obtained. The proposed GP-DMF method attempts to provide a generalized data-orientation multi-sensor measurement method which does not rely on the sensor itself and this makes it having potential to be used in a wide application fields.

GAUSSIAN PROCESS BASED DATA MODELLING AND FUSION METHOD

The proposed data modelling and fusion method is shown in FIG. 1. The measurement datasets from different sensors are firstly modelled with Gaussian Process (GP) modelling. Hence, the mean surfaces and their associated uncertainties are determined. The mean surfaces are registered into a common coordinate system with the iterative closest point (ICP) method. As a result, the two datasets are fused together using the maximum likelihood method.

The measurement process can be considered as a Gaussian process (GP) which is a stochastic process with a mean function m and a covariance function k , and it can be expressed as Eq. (1) [7].

$$f \sim GP(m, k) \quad (1)$$



FIGURE 1. Diagram of the data modelling and maximum likelihood data fusion method

In the present study, the Gaussian process modelling is undertaken by using the Gaussian processes for machine learning (GPML) toolbox [8]. The mean function is chosen to be zero mean since the underlying surface is supposed to be unknown and the covariance function is chosen to be squared exponential function since the measured surface is targeted to be continuous freeform surface.

Although the coordinate information of the sensors equipped with the multisensor CMMs can be calibrated with a standard artifact [9], there is still residual error for the relative position for the sensors. The registration process aims to minimize the residual error and the ICP [10] method is used in this study.

The mean surface and the associated uncertainty are then fused together with a Bayesian inference based maximum likelihood method. For two measurement data z_{m1} and z_{m2} , which can be determined by

$$z_{m1} = z_1 \pm \sigma_1 \quad (2)$$

$$z_{m2} = z_2 \pm \sigma_2 \quad (3)$$

where z_1 and z_2 are the measurement results and σ_1 and σ_2 are the associated uncertainties. The sensor models are given by the Gaussian likelihood function as shown below:

$$p(x | z_1, \sigma_1^2) = \frac{1}{\sigma_1 \sqrt{2\pi}} \exp\left(-\frac{(x - z_1)^2}{2\sigma_1^2}\right) \quad (4)$$

$$p(x | z_2, \sigma_2^2) = \frac{1}{\sigma_2 \sqrt{2\pi}} \exp\left(-\frac{(x - z_2)^2}{2\sigma_2^2}\right)$$

Hence,

$$p(x | z_1, \sigma_1^2, z_2, \sigma_2^2) = p(x | z_1, \sigma_1^2) p(x | z_2, \sigma_2^2) \quad (5)$$

$$= \frac{1}{2\pi\sigma_1\sigma_2} \exp\left(-\frac{(x - z_1)^2}{2\sigma_1^2} - \frac{(x - z_2)^2}{2\sigma_2^2}\right)$$

The log of above function is:

$$L(x | z_1, \sigma_1^2, z_2, \sigma_2^2) = C_1 - \left(\frac{(x - z_1)^2}{2\sigma_1^2} + \frac{(x - z_2)^2}{2\sigma_2^2}\right) \quad (6)$$

From Bayes' theorem, the fused best estimate is given by

$$\hat{z} = \arg \max [L(x | z_1, \sigma_1^2, z_2, \sigma_2^2)] \quad (7)$$

$$\text{Hence, } \frac{\partial L}{\partial x} = 0$$

The fused result can be derived by Eq. (8) while Eq. (9) shows the standard deviation which represents the fused uncertainty.

$$\hat{z} = \left(\frac{z_1}{\sigma_1^2} + \frac{z_2}{\sigma_2^2}\right) / \left(\frac{1}{\sigma_1^2} + \frac{1}{\sigma_2^2}\right) \quad (8)$$

$$\hat{\sigma} = 1 / \sqrt{\left(\frac{1}{\sigma_1^2} + \frac{1}{\sigma_2^2}\right)} \quad (9)$$

Eq. (8) and Eq. (9) show that the fused data represent the overall best estimated measurement result with a lower uncertainty than both original data. The fusion of the two measurement data is illustrated in Fig. 2. Eq. (8) and Eq. (9) also show that when one measurement is more accurate than another one, the weighting for the more accurate one is much larger than the other one in a quadratic relation. When the measurement uncertainty for the less accurate sensor is more than three times of the accurate one, the influence of the less accurate one is insignificant for the overall result.

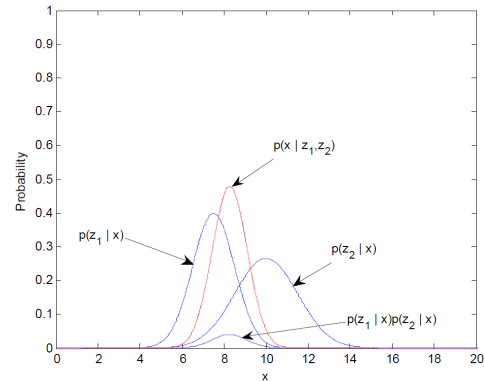


FIGURE 2. Fusion of two Gaussian distributions [11]

EXPERIMENT VERIFICATION WITH SIMULATED SURFACES

Two simulated sinusoidal surfaces with different levels of measurement noises are designed which are determined by Eq. (10).

$$z = \sin\left(\frac{\pi}{10}x\right) + \cos\left(\frac{\pi}{10}y\right) + N_m \quad (10)$$

where N_m is the measurement noise, for data set 1, $N_m = 0.01$ mm, for data set 2, $N_m = 0.005$ mm.

FIG. 3 and FIG. 4 show the mean surfaces and the estimated uncertainties of the data set 1 and data set 2 after Gaussian Processing modelling. The result shows that the simulated measurement noise is well modelled by the Gaussian Process in the covariance.

After the Gaussian Process data modelling, the two datasets are then fused with the maximum likelihood method and the result is shown in FIG. 5. The result shows that the fused uncertainty value is smaller than both original measurement data. The

uncertainty for dataset 1 is 10 μm while the uncertainty for dataset 2 is 5 μm , which is much smaller than dataset 1. The fused data is 4.5 μm which only shows small improvement than the dataset 2. The deviation from fused mean surface to design surface is shown in FIG. 6 and the result shows that the deviation is well covered by the uncertainty coverage in the fused data.

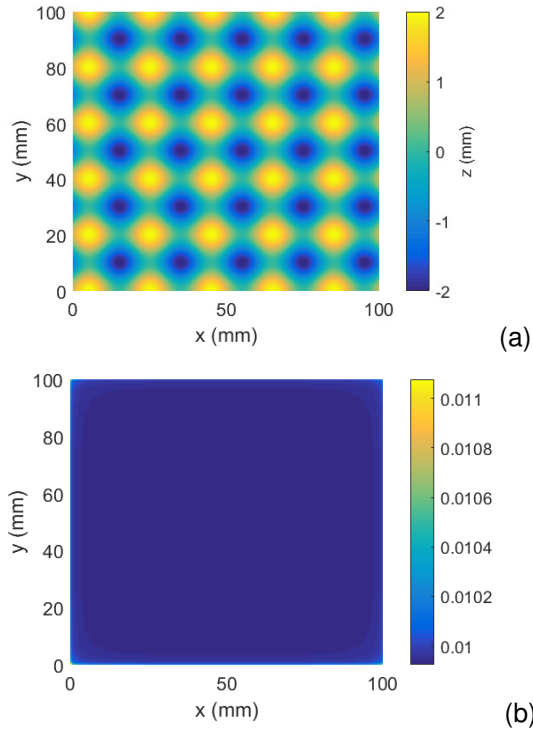


FIGURE 3. Mean and uncertainty of dataset 1

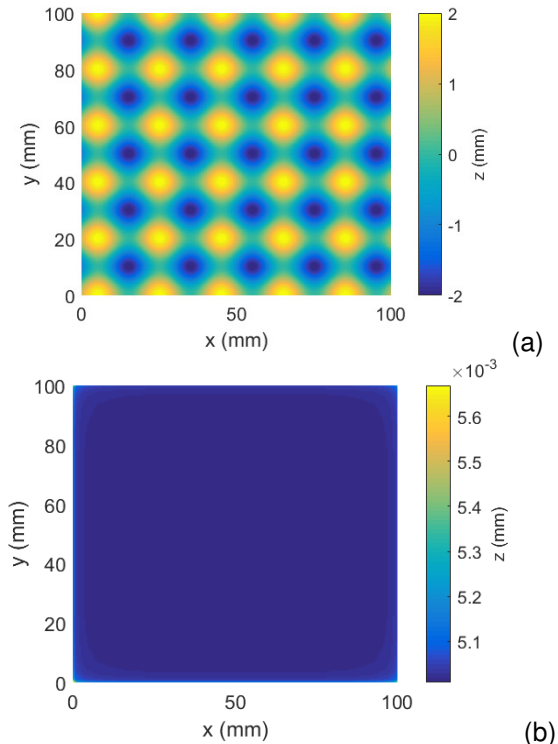


FIGURE 4. Mean and uncertainty of dataset 2

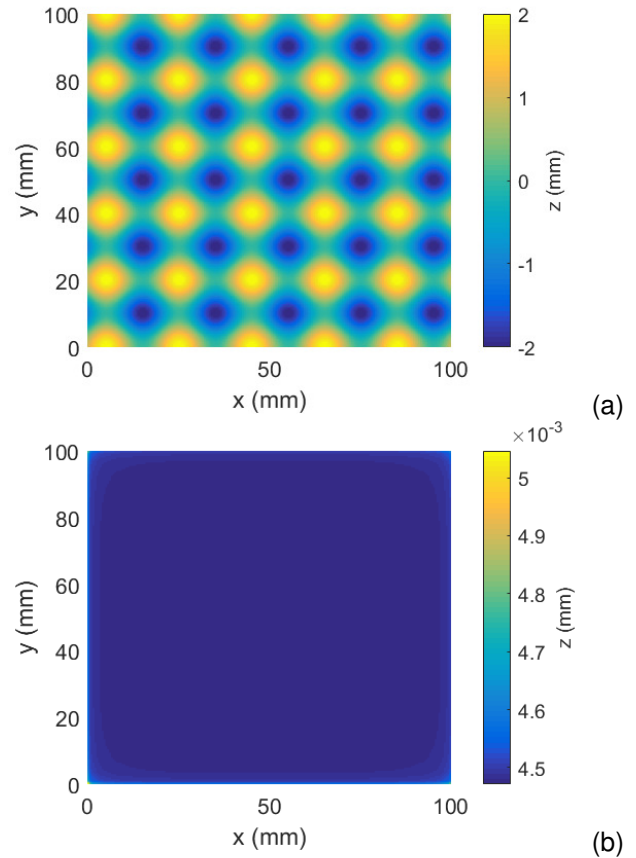


FIGURE 5. Fused mean surface and uncertainty.

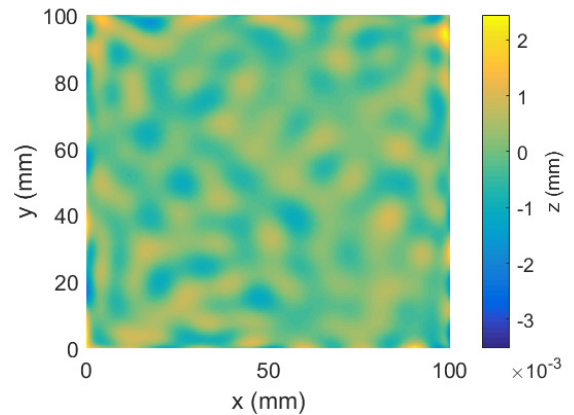


FIGURE 6. Deviation from fused mean surface to design surface.

EXPERIMENTAL VERIFICATION WITH A MULTISENSOR CMM MACHINE

To verify the proposed GP-DMF method, a series measurement experiments were conducted on a Werth multisensor CMM machine. A workpiece was designed, machined, measured and followed by the data processing procedure. The workpiece was designed to be a sinusoidal surface and it was machined by a CNC machine. The machine surface was measured by a laser sensor and a trigger probe. FIG. 7 shows the measurement process using the laser sensor of the CMM machine.

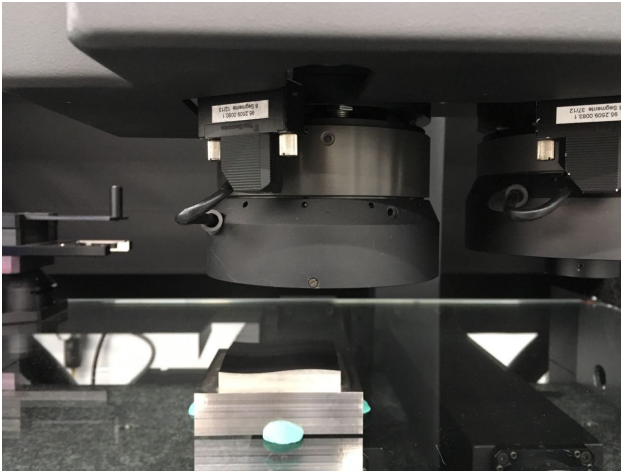


FIGURE 7. Measurement process using the laser sensor

The measurement starts with the laser sensor and a series sampling points were obtained as shown in FIG. 8.

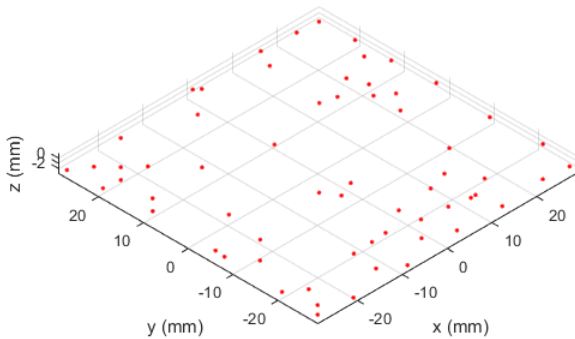


FIGURE 8. Sampling points using laser sensor

The mean surface and the estimated measurement uncertainty for the measured dataset using Gaussian process are shown in FIG. 9. It is interesting to note that the measurement uncertainty for the laser sensor is quite large (from 26 to 32 μm) which may be caused by the surface characteristics such as reflection.

A CAD model was generated as shown in FIG. 10 by using the mean surface of the laser scanned data so as to guide the measurement of the trigger probe. The sampling data using the trigger probe is shown in FIG. 11. The mean surface and the estimated uncertainty for the measurement data of the trigger probe after Gaussian process are shown in FIG. 12. The results show that the uncertainty range is about 6-8 μm which is much smaller than that from the laser sensor. FIG. 13 shows the mean surface and uncertainty after data fusion, and it shows that there is only a small amount of improvement in the fused uncertainty..

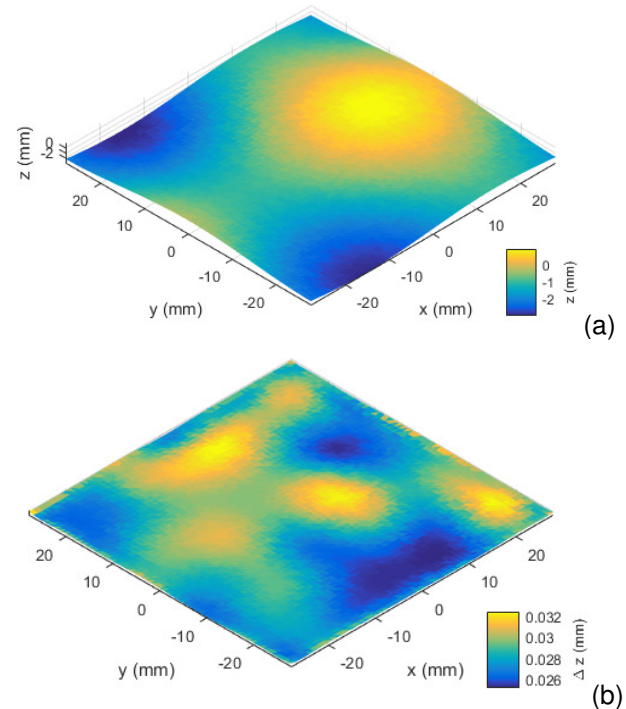


FIGURE 9. Mean surface and estimated uncertainty for the measured data from laser sensor

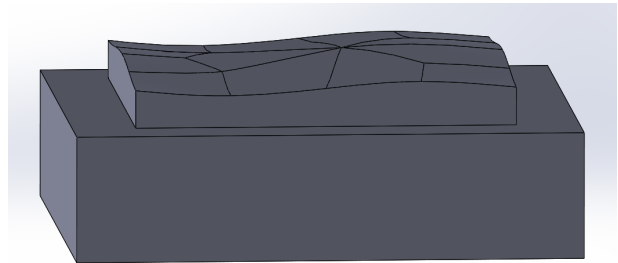


FIGURE 10. CAD model generated from the mean surface of the measured data of the laser sensor

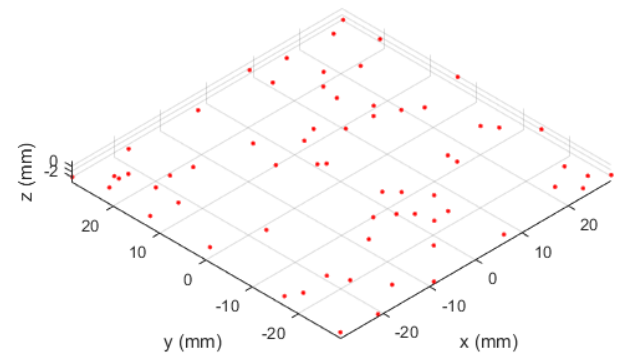


FIGURE 11. Sampling points using trigger probe

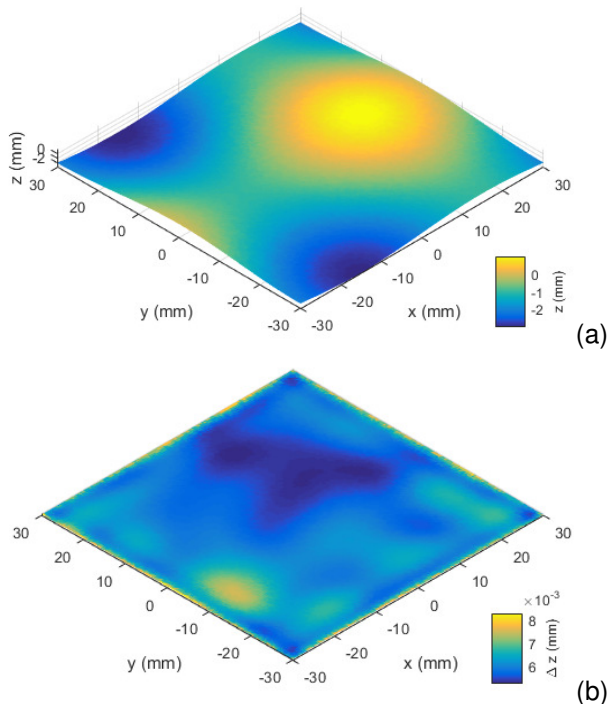


FIGURE 12. Mean surface and estimated uncertainty for the data from trigger probe

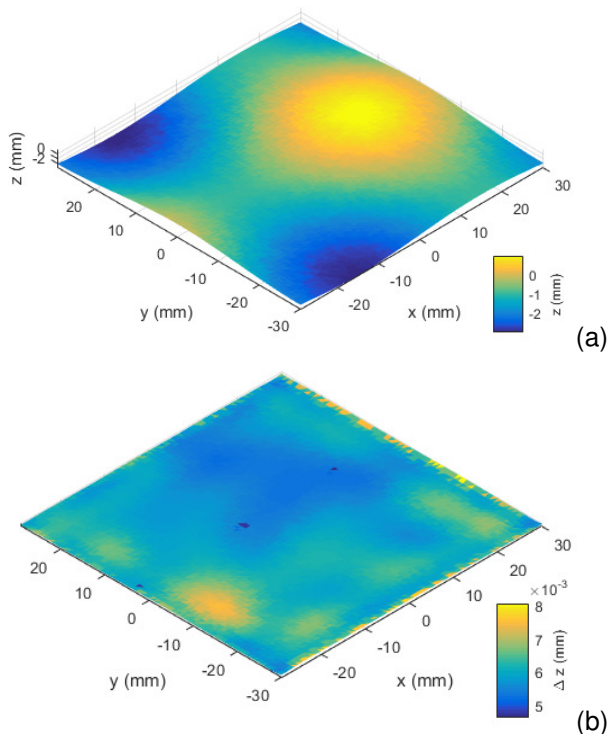


FIGURE 13. Fused mean surface and estimated uncertainty

CONCLUSION

This paper presents a Gaussian process based data modelling and fusion (GP-DMF) method and the effectiveness of the method is verified in a simulation and real measurement experiment. The result shows that the fused result has a lower

measurement uncertainty. The proposed method is technically feasibility to be used to enhance the measurement ability for multisensor CMMs.

ACKNOWLEDGMENT

The work was mainly supported by the Research Grants Council of the Government of the Hong Kong Special Administrative Region (Project no.: No.15202814). The work was also supported by a PhD studentship (project account code: RTHC) from The Hong Kong Polytechnic University.

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CHARACTERIZATION AND EVALUATION OF INVOLUTE GEAR FLANK DATA USING AN AREAL MODEL

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INTRODUCTION

Involute gears are decisive components in power transmission systems [1]. The geometry of a gear must be designed and manufactured to targeted tolerances to increase lifetime, reduce vibration and noise emission during operation [2]. For a better functional performance, the form of a gear flank is intentionally deviated from the ideal involute surface. Such deviations are known as flank modifications [5]. Modern gear manufacturing technology is capable of producing desired geometrical features on a gear flank, but the current quality control of those features is still based line features (profile and helix/lead), which are measured and graphically analyzed to assess the quality of the entire gear according to [3,4]. The measurands on gear flanks are limited to those predefined two-dimensional (2-D) lines, even though the complete geometry of a gear flank determines its functional properties.

FIGURE1 shows the setup of a helical gear being measured on a Leitz PMMF302010 coordinate measuring machine (CMM), where an eight stylus star probe is used to access all teeth and gaps on the gear. It takes nearly five minutes to measure four teeth (left and right flanks, profile and lead line) and the pitch deviations of all teeth besides the time spent on clamping and alignment.



FIGURE1. Setup of a classical gear measurement on a CMM using an eight stylus start probe.

The measured gear line features are converted to deviation charts, which are tolerated and evaluated in two dimensions. Gear deviation parameters are calculated according to established evaluation standards (e.g. [4]) as illustrate by the example in FIGURE 2. These parameters can either be used to improve the gear manufacturing processes and/or to predict the performance of a gear pair during operation.

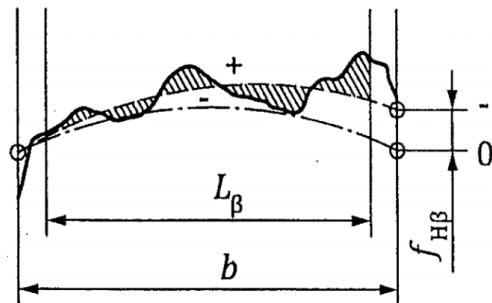


FIGURE 2. A typical evaluation chart of a measured gear helix/lead with crowning modification [4].

This standardized evaluation method has been used in gear production for decades, most commonly in automobile industry. Here, the measurement of only four teeth on the whole gear is acceptable because of very short machining times.

However, as the size of the gear increases (diameters up to several meters), the whole gear flank deviation cannot be represented by those "line features" on a few teeth. This is due to the variation of machine conditions and environmental factors during hours of manufacturing processes, which imposes various deviations on the individual tooth. Instead of lines, the topography of the whole flank surface should be measured and evaluated to represent the quality of a gear.

This paradigm shift requires the advancement on both sensing techniques and data processing methodologies.

In this paper, a 3-D gear flank model is introduced, derived from the classical 2-D involute equation [7]. With this model, the 3-D flank topographic deviation in the normal direction can be extracted from the measured points, which is known as plumb line distance. The topographic deviation, also referred to as areal deviation, is then treated as a bivariate height function which can be decomposed into the sum of a series of orthogonal polynomials. The coefficients of the orthogonal polynomials are used to reconstruct the deviation map and to calculate gear flank modifications parameters and/or form deviations parameters.

MODELS OF 2D INVOLUTE CURVE AND 3D INVOLUTE FLANK

In a Cartesian coordinate system, the fundamental equation of a 2-D involute curve is given by:

$$\begin{cases} x = r_b \cdot [\cos(\xi + \Lambda) + \xi \cdot \sin(\xi + \Lambda)] \\ y = r_b \cdot [\sin(\xi + \Lambda) - \xi \cdot \cos(\xi + \Lambda)] \end{cases} \quad (1)$$

where r_b , ξ and Λ are the radius of base circle, rolling angle and initial angle of the involute curve, respectively. The gear flank profile is formed by a certain range of ξ . The value of Λ depends on the choice of the involute's origin.

An arbitrary point on the involute curve could be represented in a Cartesian coordinate system or a cylindrical coordinate system as shown in FIGURE 3.

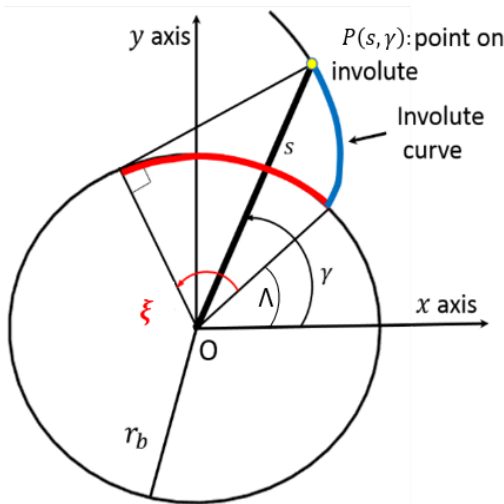


FIGURE 3. Involute curve generated by the rolling angle, ξ , on a base circle with radius r_b , in a cylindrical coordinate system (s, γ).

For the 3-D model of a helical involute gear ($\beta_b \neq 0$), the flank can be generated by “stacking” layers of transverse planes with 2-D involutes, where the involute origin follows a thread on the base cylinder. This can be mathematically described by associating the initial angle Λ to the third dimension z [7]:

$$\begin{cases} x = r_b \cdot [\cos(\xi + \Lambda(z)) + \xi \cdot \sin(\xi + \Lambda(z))] \\ y = r_b \cdot [\sin(\xi + \Lambda(z)) - \xi \cdot \cos(\xi + \Lambda(z))] \\ z = z \end{cases} \quad (2)$$

where $\Lambda(z) = \Lambda_0 + z \cdot \tan \beta_b / r_b$ is a function of the helix angle β_b on the base circle and the initial angle on the gear's bottom transvers plane Λ_0 .

PLUMB LINE DISTANCE AND AREAL DEVIATION ON INVOLUTE FLANK

According to the gear meshing principle, the contact force between two gear tooth flanks is in the surface normal direction. Undesired artifacts that are deviations from the designed geometry in the surface normal direction influence the gear's functional performance. Therefore, the metrological characteristics of a gear flank should be directly measured or calculated in the surface normal direction.

The relationship between a measured point to the corresponding nominal point and the gear flank deviation can be modeled as the vector summation:

$$\vec{p}_{meas} = \vec{p}_{nom} + \vec{d}_{lot} = \vec{p}_{nom} + |\vec{d}_{lot}| \vec{n} \quad (3)$$

where $\vec{p}_{meas}$ is the vector of the measured point, $\vec{p}_{nom}$ is the vector of the corresponding nominal point and $\vec{d}_{lot}$ is the deviation vector pointing from the nominal point to the measured point in the surface normal direction, respectively. The vector length, $|\vec{d}_{lot}|$ is the plumb line distance. $\vec{n}$ is the unit normal vector. FIGURE 4 shows the 3-D model of a gear tooth in a Cartesian coordinate system, illustrating equation (3).

Points on the tooth flank can be described by the involute equation (2) in a Cartesian coordinate system. In addition, to characterize the surface topography, flank deviation is described by a pair of local surface coordinates and the corresponding plumb line distance, (u, v, d_{lot}). The u axis and v axis follow along the profile and

helix/lead direction, respectively, as shown in FIGURE 4.

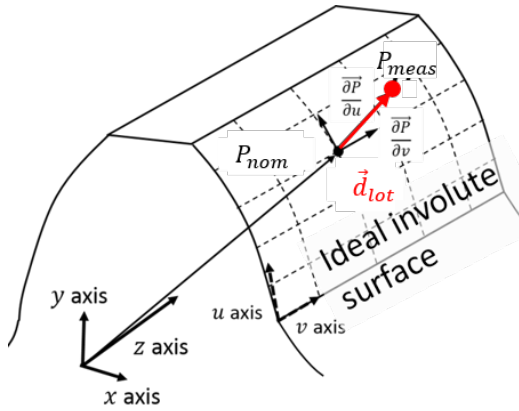


FIGURE 4. Plumb line distance in 3-D Cartesian system and surface coordinates (u, v) on flank.

The plumb line distance can be obtained by solving the vector equation (4) for a measured point given in cylindrical coordinate system ($s_{meas}, \gamma_{meas}, z_{meas}$) [7]:

$$|\vec{d}_{tot}| = \frac{r_b}{\sqrt{1+\tan^2 \beta_b}} \left\{ \sqrt{\frac{s_{meas}^2}{r_b^2} - 1} - \operatorname{atan} \left(\sqrt{\frac{s_{meas}^2}{r_b^2} - 1} \right) - \gamma_{meas} + \eta_b + z_{meas} \frac{\tan \beta_b}{r_b} \right\} \quad (4)$$

Collecting the deviation at each measured point, an areal deviation map is formed representing the topography of the entire measured flank. FIGURE 5 shows a simulated ($x_{meas}, y_{meas}, z_{meas}$) 3-D point cloud (left) and the calculated areal deviation map (right).

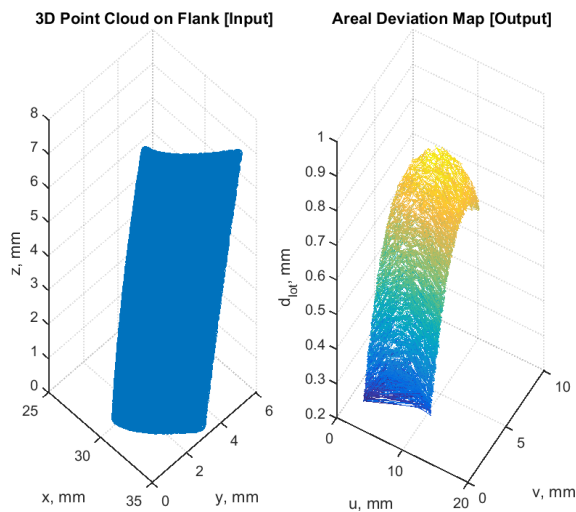


FIGURE 5. A 3D point cloud of the simulated “measured” gear flank (left) and its areal deviation map (right) calculated by the plumb line distance equation.

MODIFIED INVOLUTE FLANK

Flank modifications are desired alterations applied to the basic involute helicoid [3], intentionally introduced in both profile and helix/lead direction. The depth of a modification varies from micrometers to tenths of a millimeter depending on the size and application of a gear.

FIGURE 6 illustrates four fundamental gear flank modifications. It is in the surface normal direction of the basic involute flank that a modification is mathematically defined. Therefore, the modification information is inherently contained in the areal deviation map obtained by equation (4).

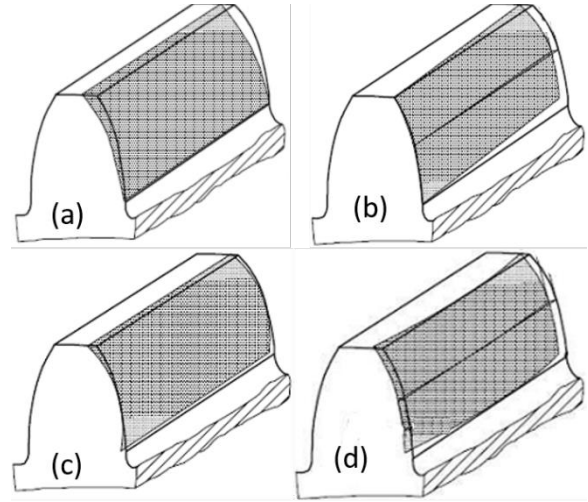


FIGURE 6. Typical gear flank modifications. a) 1st order profile slope, b) 1st order helix/lead slope, c) 2nd order profile crowning, d) 2nd order helix/lead crowning [3].

All of the mentioned modifications are independently described in profile or helix/lead direction. They are superimposed to the basic involute flank to form a modified flank. For example, a profile line with slope and crowning modifications can be mathematically described as a combination of the 1st and 2nd order polynomials:

$$A(u) = -\frac{4C_\alpha}{L_u^2} u^2 + \frac{4C_\alpha + C_{H\alpha}}{L_u} u \quad (5)$$

where, C_α , $C_{H\alpha}$, L_u , u are the amount of profile crowning, the amount of profile slope, the extended evaluation range [3] and the surface coordinate, as shown in FIGURE 7.

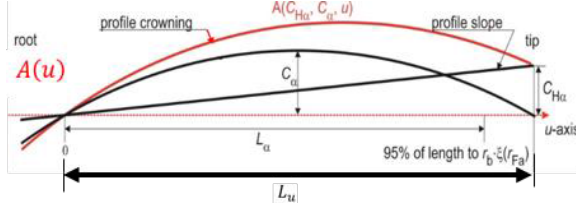


FIGURE 7. Profile modification as a function of surface coordinate u .

2D CHEBYSHEV POLYNOMIALS FOR FLANK RECONSTRUCTION

In optical interferometry, the measured wavefront of an optic can be decomposed into a 2-D Chebyshev polynomial on a rectangular aperture [8]. The 2-D Chebyshev polynomials are used in this paper to reconstruct the gear flank and to calculate the values of areal deviations and modification parameters.

In 2-D space, the Chebyshev polynomial terms denoted as $C_n(x, y)$ are a set of strictly orthogonal basis function defined on the square domain of $[-1, 1] \times [-1, 1]$. They are generated by multiplying two 1-D Chebyshev terms:

$$C_n(x, y) = T_p(x)T_q(y) \quad (6)$$

with p and q representing the orders along the respective axes. The orthogonality of the 2-D Chebyshev polynomials can be described by:

$$\int_{-1}^1 \int_{-1}^1 \frac{C_i(x, y)C_j(x, y)dxdy}{\sqrt{1-x^2}\sqrt{1-y^2}} = \begin{cases} 0: i \neq j \\ K: i = j \end{cases} \quad (7)$$

where the normalization factor K is determined by the values of p and q in equation (6):

$$K = \begin{cases} \pi^2, p = q = 0 \\ \frac{\pi^2}{4}, p \neq 0 \text{ and } q \neq 0 \\ \frac{\pi^2}{2}, \text{else} \end{cases} \quad (8)$$

FIGURE 8 shows the first nine terms of the 2-D Chebyshev polynomials $C_n(x, y)$ by re-arranging the term index n starting from $p = q = 0$.

Given the measured flank deviation map $W(x, y)$, a finite number of 2-D Chebyshev terms $C_n(x, y)$ can approximate the geometry of $W(x, y)$ as $W'(x, y)$:

$$W(x, y) \approx W'(x, y) = \sum_{i=0}^N A_i C_i(x, y) \quad (9)$$

where A_i is the coefficient for the i th term $C_i(x, y)$ of the 2-D Chebyshev polynomial and N is the total number of terms used for reconstructing $W'(x, y)$.

A_i is calculated by evaluating the double integral over the domain of $[-1, 1] \times [-1, 1]$:

$$A_i = \frac{1}{K} \int_{-1}^1 \int_{-1}^1 \frac{W_i(x, y)C_i(x, y)dxdy}{\sqrt{1-x^2}\sqrt{1-y^2}} \quad (10)$$

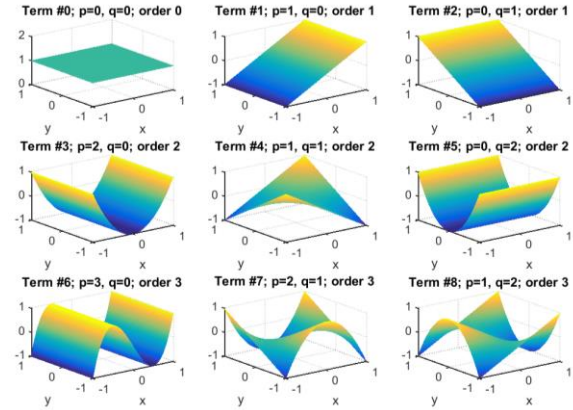


FIGURE 8. First nine terms of 2D Chebyshev polynomials $C_0(x, y)$ to $C_8(x, y)$.

The geometric information of the modifications shown in FIGURE 6 mathematically resemble the 1st, 2nd, 3rd and 5th order of the 2-D Chebyshev terms as shown in FIGURE 8. An unknown areal flank data can be decomposed into the sum of a series of 2-D Chebyshev terms, the A_i coefficients can be converted to the corresponding values of the modification parameters.

VERIFICATION BY SIMULATED FLANK DATA WITH MODIFICATIONS

A gear flank with simulated modifications generated by the spatial coordinate equation (2) and the modification equation (5) was used to test the proposed Chebyshev method.

The point cloud was selected as 250 by 250 points to preserve the orthogonality of the 2-D Chebyshev terms. The coefficients for the first 35 terms are displayed in FIGURE 10. 2D Chebyshev spectrum of the input areal map data, representing the 2-D Chebyshev spectrum of the input areal map data. The coefficient of the 0th order component, A_0 , is nonzero, which means that the whole actual flank is translated by a constant distance in the surface normal direction. This component can therefore be interpreted as

the pitch deviation of the tooth. A_0 to A_5 are applied in calculation of surface parameters, which are shown as the approximated values in TABLE 1.

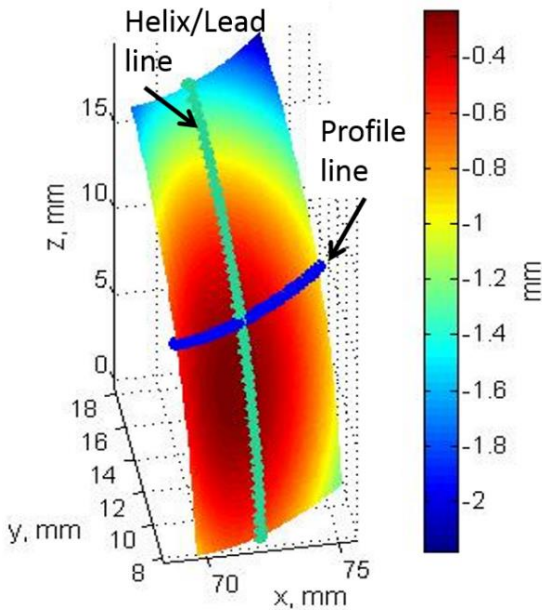


FIGURE 9. 3D gear data with simulated modifications (the colors represent the values of plumb line distances with respect to the ideal involute surface).

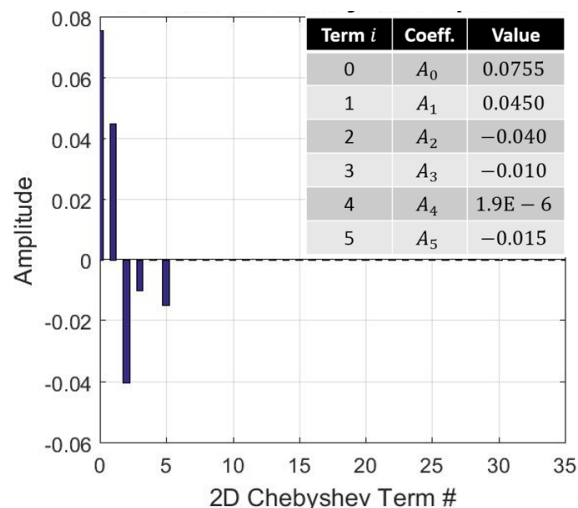


FIGURE 10. 2D Chebyshev spectrum of the input areal map data.

The approximated parameters retrieve the designed parameters with a relative error of no more than 0.1%.

TABLE 1. Values of the simulated and approximated modification parameters.

Parameter	Definition	Simulated [μm]	Approximated [μm]
$C_{H\alpha}$	Profile slope	90	90.068
C_α	Profile crowning	20	20.019
$C_{H\beta}$	Helix slope	80	80.052
C_β	Helix crowning	30	30.030

CONCLUSION AND FUTURE WORK

This paper presents a new evaluation method of areal gear flank deviations obtained as plumb line distances. Simulations shows that the designed flank modifications are obtained from the term coefficients of the 2-D Chebyshev polynomials with acceptable relative error. This method opens opportunities for a next generation gear evaluation technique when massive areal flank data, obtained e.g. by optical sensor are prevalently available in industry.

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Diamond Machining of a 14th Order Alvarez Lens for IR Applications

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INTRODUCTION

Traditionally, optics have been limited to flats or axi-symmetric spherical shapes due to the relative ease of manufacturing. However, single crystal diamond milling and coordinated axis turning allow for near arbitrary surface topographies, freeform optics [1], to be machined into various optical materials. Building on the diamond turning literature [2,3], this paper reports on the development of machining parameters for a brittle IR transparent chalcogenide glass (IRG 26) [4] with minimal surface fracture. Parameters were developed for both freeform three-axis milling and coordinated axis turning, and the results were applied to the manufacture of a 14th order polynomial freeform surface: an Alvarez lens [5]. The form and finish of the freeforms were measured using a ZYGO scanning white light interferometer (SWLI). The lenses were functionally tested by measuring focal length measurements as a function of lens shear as per the Alvarez lens effect.

OPTICAL DESIGN

Building from the work of Smilie, et al. [6], a 14th order polynomial Alvarez lens was designed:

$$Z_F(x, y) = \sum_{k=1}^n \left(\frac{1}{2} a_{2k} (x^2 + y^2)^k \right) + \sum_{k=1}^n (b_{2k} \int (x^2 + y^2)^k dx) + c_0 x \quad (1)$$

The coefficients for Equation 1 are shown in Table 1. This design uses As₄₀Se₆₀ chalcogenide glass (trade name IRG 26) as the lens material and has an operating

wavelength of 4 μ m. A stacked Alvarez lens pair acts as a plano-convex lens for a positive shift in the x direction. A plano-concave lens is formed when sheared in the negative x direction. Higher order terms in addition to the more well-known cubic polynomial Alvarez lens help to reduce various optical aberrations and shift the focal length to a finite value at zero x shift while retaining the composite lens properties. This lens design has a clear aperture diameter of 14.5 mm and enables focal length variation from 135 mm to 35 mm over a ± 1 mm shearing displacement in the x-direction.

Table 1: 14th Order Freeform Coefficients
Freeform surface coefficients ($c_0 = 0.0262$)

Order (k)	a_{2k} (mm^{-2k+1})	b_{2k} (mm^{-2k})
1	-3.62E-03	-1.01E-03
2	8.50E-08	9.45E-08
3	-4.00E-12	-1.78E-11
4	2.88E-16	4.39E-15
5	-1.61E-18	-7.61E-18
6	2.31E-20	9.49E-20
7	-1.31E-22	-5.34E-22

PARAMETER

DEVELOPMENT

All work was performed on a Moore Nanotechnology 350 FG ultra-precision machining center with a 10 krpm main spindle and a 60 krpm auxiliary milling spindle.

As demonstrated in Troutman et al. [7,8], IRG 26 exhibits a dramatic change in cutting behavior during orthogonal turning for uncut chip thicknesses exceeding approximately 600 nm. This is characterized by an abrupt decrease in cutting process forces and also by increased surface roughness and reduced reflectivity due to the presence of a pitted, fractured surface. While the cutting behavior under simplified conditions is well understood, existing machining models are not sufficient to predict more complex machining scenarios from orthogonal cutting data alone [9]. Therefore, the best approach for selecting cutting parameters is to use intuition from simplified experiments to estimate parameter ranges and then perform targeted empirical testing.

Coordinated axis turning

In round-nosed turning, uncut chip thickness varies around the tool nose and is controlled by feed rate and depth of cut. To determine acceptable turning parameters, bands were face-turned on a specimen using a zero-rake single crystal diamond turning tool with nose radius of 1.50 mm. Cutting velocity and depth of cut were held constant at 2 m/s and 5 μm , respectively, and feed rate was varied from 0.3 $\mu\text{m}/\text{rev}$ to 10 $\mu\text{m}/\text{rev}$. The specimen was examined using SWLI and surface finish of 5 nm Sa was achieved with appropriate parameters. Increased surface fracture was present at feed rates above 10 $\mu\text{m}/\text{rev}$.

While rake angle is constant during conventional turning, localized effective rake angle is a function of both part and tool geometry during slow-tool-servo operations. Slope analysis of the Alvarez surface indicated that for a zero-rake tool, effective rake angle would vary between $\pm 1.5^\circ$. It is well-known that negative rake angles are more favorable in machining of brittle materials. To examine the effect of slightly positive rake angles on cutting behavior, a sine-wave test specimen was machined. One the sine specimen the effective rake angle cycled between $\pm 3^\circ$ three times per

revolution. After machining, regions corresponding to the different rakes were examined using SWLI. Figure 1 illustrates the sine specimen and subsequent analysis. Small positive rake angles were not found to significantly affect cutting mechanics.

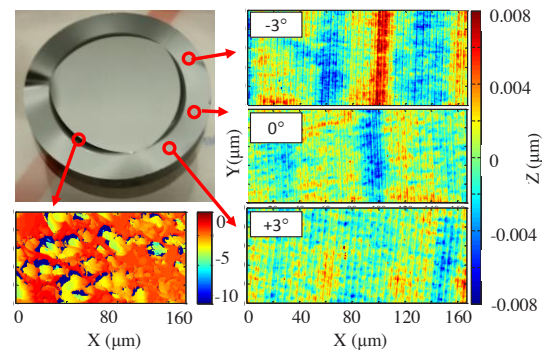


FIGURE 1: Clockwise from top-left: sine wave specimen, SWLI of regions of -3° , 0° , $+3^\circ$ effective rake, and brittle-fractured shoulder region for comparison.

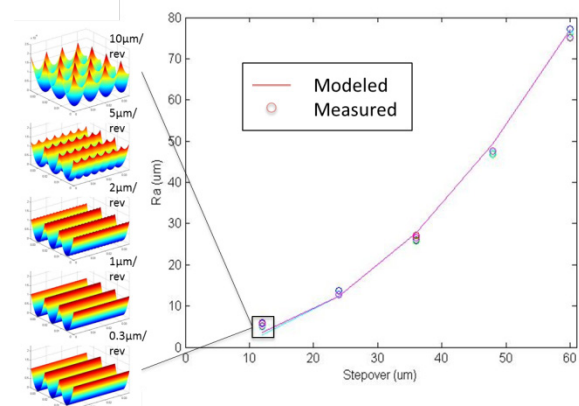


FIGURE 2: Predicted Sa compared to Measured. The surface plots show the expected surface profiles for the range of feedrates. It should be noted that the main dictator of surface finish is the finishing step-over.

Freeform Milling

During the single crystal diamond ball milling process the rake angle remains constant. However, as described in [9], instantaneous uncut chip thickness varies around the tool nose is controlled by feed rate, step-over, and depth of cut. Further, the spindle is often inclined relative to the Z axis to prevent low surface velocities. To determine milling parameters in IRG 26 that

produce a surface with minimal surface fracture, a grid of parameters was evaluated at each of three spindle inclination angles: 0°, 22.5°, and 45°. Experiments were performed in a single specimen of IRG 26 using a single-flute, zero-rake single crystal diamond endmill with a corner radius of 1.0 mm. The feedrates that were evaluated are 0.3, 1, 2, 5, and 10 $\mu\text{m}/\text{rev}$. The step-over values evaluated were 12, 24, 36, 48, 60 μm and the depth of cut was 200 μm . Surfaces were measured with SWLI. Figure 1 shows the predicted S_a compared to the measured S_a for each parameter set. A surface finish of less than 5 nm S_a was attained for some parameters.

MANUFACTURING

Two sets of the Alvarez lenses were manufactured, one with coordinated axis diamond turning and one with freeform raster milling. Unlike macroscale applications, conventional computer aided manufacturing (CAM) software is unsuitable for optics with sub-micrometer form and finish tolerances. Instead, toolpath for both machining methods was generated using a custom MATLAB® CAM suite which functions analytically based upon the surface prescription. Toolpath generation follows this algorithm, which can be adapted as needed to any machining configuration:

1. The optical prescription is translated into a coordinate system which most closely matches the machining process (i.e. cylindrical coordinates for coordinated axis turning, rectangular coordinates for milling).
2. A list of raw coordinates on the surface to be machined are selected, defining a toolpath spiral or raster pattern across the part.
3. The normalized outward normal vector is computed at each point on the surface.
4. The raw coordinates are offset along the outward normal by the tool nose radius (turning) or corner radius (milling).

As described in Troutman et al. [8,10] for the case of coordinated axis turning, this toolpath generation strategy was extended to compensate for repeatable error sources using an artifact-based approach for both machining configurations.

Coordinated axis turning

For coordinated axis turning, a tilted flat setup artifact was used to map tool waviness before machining of the optics. Measurement of the setup artifact on a Fizeau interferometer and subsequent analysis of slope-dependent errors indicated the tool waviness trend. By modeling slope-dependent nose radius deviation, step 4 of the toolpath generation strategy was extended to compensate for the errors.

Rough machining was performed with a 25 μm depth of cut, 25 μm feed per revolution, and a light mineral oil mist. Finish machining was conducted with a 5 μm depth of cut, a 10 μm feed per revolution, and lubrication via a thin jet of mineral oil directed through a hypodermic needle towards the cutting zone. The outer diameter of the optic and the mounting flange were diamond turned, serving as physical datums. A small orientation flat was also milled on mounting flange for metrology purposes.

Freeform raster milling

For freeform raster milling, a concave spherical setup artifact was utilized. In addition to identification of tool waviness via slope-dependent errors, identification of ogive-shaped error trends were used to determine tool diamond decenter. Following a slightly modified auxiliary strategy, the toolpath can be compensated:

1. The outward normal vector component in only the X-Y plane is determined and is normalized.
2. Each toolpath point is offset along the X-Y normalized vector by the decenter distance.

Rough machining was performed with two passes at 300 μm depth of cut, 10 μm feed

per revolution, and a light mineral oil mist. A semi-finish pass was then used at 100 μm depth and 10 μm feed per revolution. Finish machining was conducted with a 100 μm depth of cut, a 2.5 μm feed per revolution, and lubrication via a thin jet of mineral oil. Measurement fiducials in the form of three balls milled into the flange of the optic allow the optical surface coordinate system to be located for metrology.

METROLOGY

Coordinated axis turning

Surface finish of the optic was measured using SWLI. Under a 50x objective, the RMS surface finish after low pass filtering with a 15 μm cutoff was approximately 6 nm. Form was evaluated with a 5x objective using an automated stitching algorithm. This objective was chosen to minimize the number of stitched maps. Due to the acceptance angle of this objective, small portions of the surface where local slope exceeded approximately 2.5° were not measurable.

Additional form verification was performed on an OptiPro UltraSURF noncontact measuring instrument using a chromatic confocal probe. As shown in Figure 3, both instruments agree, indicating similar form errors of $\pm 1\ \mu\text{m}$. Eliminating other possible sources for the error, it is likely that the errors are due to dynamic axis error motions during coordinated axis machining and could possibly be reduced by the machining the optic at a slower speed.

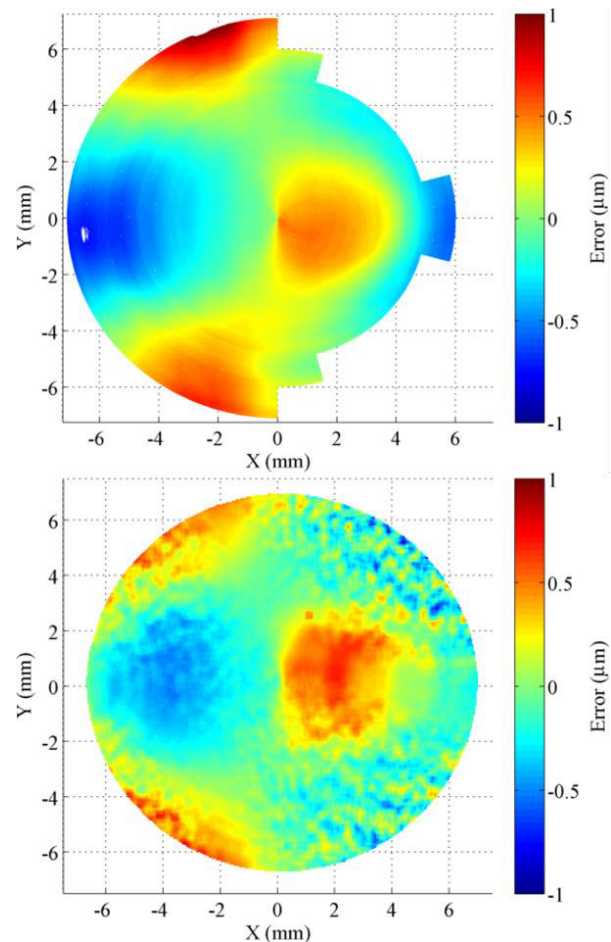


FIGURE 3: (Top) stitched SWLI measurement, (bottom) UltraSURF measurement of coordinated axis turned Alvarez lens.

Freeform raster milling

Figure 4 shows: (a) a three dimensional representation of the machined surface; (b) a validated point cloud generated by the machining code; (c) the raster milling process; and (d) the final machined IRG 26 component. One of the most versatile aspects of the milling process as demonstrated in Figure 4 (c) and (d) is the ability to machine complex fiducials (spherical seats) in the same operation as the machining of the optical surface, thus transferring the machine accuracy to the positioning accuracy of the fiducials. This aids in both metrology and assembly.

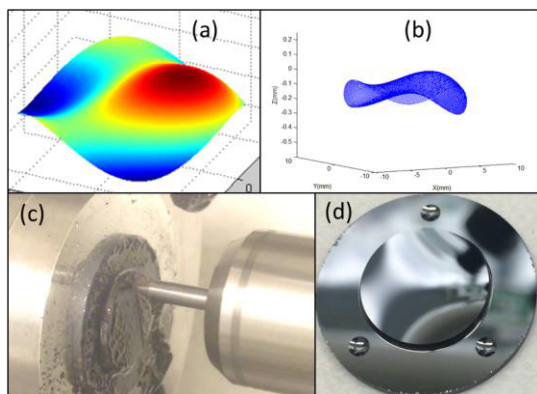


FIGURE 4: Ball milling process used to manufacture high-order Alvarez lens in IRG26 with alignment/metrology fiducials

The form and finish of the milled freeform was measured using SWLI. Finish was measured using a 50x objective and low pass filtered with a 15 μm cutoff frequency. RMS surface finish was on the order of 10 nm. Form was again evaluated with a 5x objective with an automated stitching algorithm. A slope dependent form error was identified in the measurement with a magnitude of 2 μm , as illustrated in Figure 5. It remains unclear at this time whether the errors are present in the machined surface or arise from measurement uncertainties. While the agreement between SWLI and UltraSURF shown in Figure 3 provides an independent verification of the stitching technique, this identifies an important deficit in freeform optics manufacture: the lack of independent and established off-machine metrology systems with validated uncertainty better than the expected uncertainty of the manufacturing systems. Because of the lack of adequate metrology, functional testing of freeform optic often remains the only qualitative method to validate the manufacturing process.

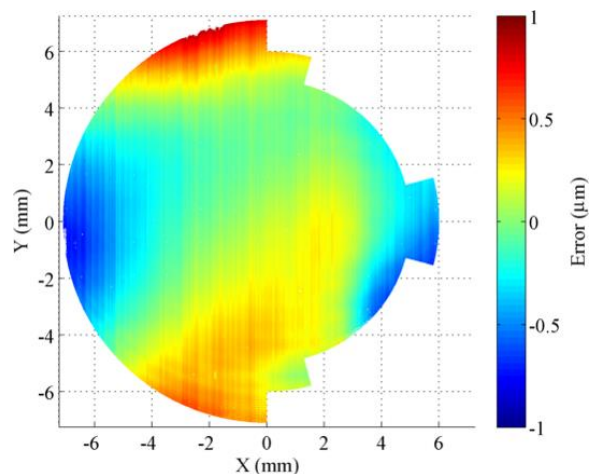


FIGURE 5: UltraSURF form error measurement of freeform raster-milled Alvarez lens.

FUNCTIONAL TESTING

To test the optical performance, a custom mounting fixture which allows micrometer level shifts in the lateral direction and subsequent variation in the power of the composite lens. The fixture was mounted on a precision mechanized stage with a positioning resolution of 10 μm . A hotplate was imaged with the detector array of an Indigo Phoenix Infrared camera. The object and camera were fixed in place. By shearing the lenses and moving the stage until the “best” image focus is attained, one can estimate the focal length using the thin lens equation. The predicted focal length matches well with that estimated from “best” focus, as shown in Figure 6. Because the “best” image focus is difficult to quantify, uncertainties are evaluated by repeated measurement.

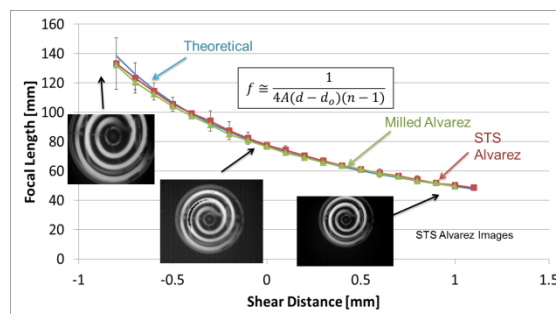


FIGURE 6: Theoretical versus measured focal length of Alvarez lens sets.

DISCUSSION

A 14th order polynomial surface was machined using ultra-precision diamond milling and by coordinated axis turning methods on a Moore Nanotechnology 350FG. Results indicate form errors of $\pm 1 \mu\text{m}$ for both the milled and turned optics. Functional testing indicates acceptable system performance and reveals no measurable performance difference between the optics manufactured by the two techniques.

ACKNOWLEDGEMENTS

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Advances in Manufacturing and Metrology of Optical Freeform Surfaces

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INTRODUCTION

Freeform optics are becoming very desirable in many applications including in space applications, beam shaping, LED lighting applications, head mounted displays, and conformal windows. Freeform optics allow for reduced optical aberrations, fewer components, and lighter weight systems. Because of the range of what a 'freeform' means, the processes and procedures for manufacturing and measuring freeforms are not standardized as with spherical optical surfaces. Because of this, it has been difficult to procure freeform optics, particularly in glass (not plastic or metal) for commercial applications in reasonable timeline with reasonable cost.

The methods developed and shown here demonstrate that it is feasible to produce to commercial grade glass freeform optics in a reasonable timeframe and cost. We will show some of the progress that we have recently made towards these goals.

FREEFORM PROCESS OVERVIEW

The general process for the production of freeform optics is surface shape generation using deterministic micro-grinding (Figure 1) and sub-aperture polishing for gray-out and figure correction. Prior to generation, an accurate surface definition is necessary generally using a solid model. During the generating and figure correction process, surface measurements are required to both evaluate the final surface figure error and to provide feedback to the manufacturing process to improve the figure error during processing.

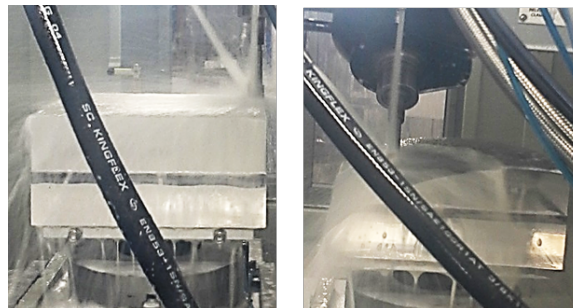


FIGURE 1: Example of the micro-grinding process on a large 330 mm square freeform optic at the start of the material removal process (left) and mid-process (right).

Sub-aperture polishing is used for both gray out (removing the damage from the generating process) and figure correction. Figure correction is generally accomplished using a dwell based approach – the polishing machine moves slower over the surface areas that are too high relative to the nominal shape causing more material removal in those areas. There are multiple types of sub-aperture polishing on the market such as Zeeko with bonnet or fluid jet polishing, QED Technologies with MRF polishing, OptiPro with the belt polishing, and OptoTech with varying polishing methods including toroid wheel polishing. Each of the methods has advantages and have shown to produce freeform optics at varying precision levels. A difficulty with these machines are their large footprint, high cost, and limited flexibility. Specifically many of these machines cannot polish a surface on a thick part (>100 mm) without an extremely large machine tool.

We have found that we needed a polishing solution that had lower cost and had increased flexibility with part size, shape, and configuration. An industrial robotic platform provides us the flexibility for varying part shapes and sizes and gives us the ability to program in varying tool paths (not provided in many commercial tools). A picture of our robotic

polisher is shown in FIGURE 2 with a large 330 mm square optic still to be polished. As shown, the robot has the flexibility to polish a large range of components and is flexible in its possible tooling. We have used varying tool shapes and sizes with varying polishing active layers, a flexibility not afforded us in many commercial

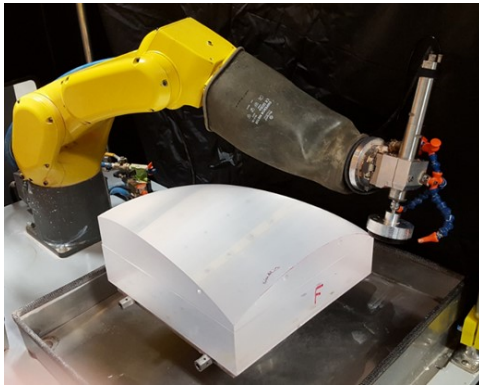


FIGURE 2: Photo of the industrial robot used for polishing with a large 330 mm square freeform optic prior to polishing.

Measurements of freeform optics is a significant area of ongoing work by many. We generally use our coordinate measuring machine (CMM) as the go-to lower accuracy ($\sim 1 \mu\text{m}$ level) tool. Higher accuracy measurements require an interferometer with a CGH (a diffractive element that produces the corresponding freeform reference wave). Other measurement options that we are actively using and developing are: stitching interferometry, a UA3P (an ultra-accurate low force CMM), or aspheric metrology equipment for freeforms with low departure.

ONGOING AREAS OF WORK

Surface Definitions

One critical area of work is to accurately produce a solid model of the surface definition. The customer may define the freeform as an equation, but transferring that equation to a solid model, which is required for generation and for the CMM, is not trivial.

Fiducials

A major difficulty with freeform optics is locating the surface in space relative to some external features. Methods for spherical and aspheric

optics (indicators on edges, air-bearing spindles and tilt/tilt stages) are not relative on optics that are not rotationally symmetric. This locating the surface in space is essential to producing a freeform that can be used in an optical system.

We use fiducials on the optic itself to establish datums on each machine in the manufacturing process. This probing is shown in FIGURE 3 and FIGURE 4 on two different types of fiducials – a flat machined on the optic's edge and a cone cut into the optic itself. Optimum methods would be to use the same fiducials used in the optic mounting in the final assembly.

FIGURE 3 and FIGURE 4 show the robotic polishing tool locating two different types of fiducials: a flat on the optic edge establishing a planar datum and a cone fiducial establishing a point. Locating the optic in the robot's coordinate frame allows for accurate polishing – it is known where to place the polishing tool to achieve a consistent hit across the surface.



FIGURE 3: Probing a flat fiducial to establish a planar datum on the robot polisher.



FIGURE 4: Probing a cone fiducial to establish a set of datums on the robot polisher.

Because the fiducials are also used to establish a part coordinate system during the measurement process on the CMM, the surface figure error of the optic can be presented in two ways. The error can be shown relative to the datums, which includes all possible alignment errors. The error could also be shown 'best fit' to the surface which always the alignment errors to be removed. The required measurement depends on the optic system and how misalignment can be handled during the assembly process.

In the following example, we manufactured the toroidal like optic (concave and convex) shown in FIGURE 5. The optic is 102 mm by 152 mm and approximately 3 mm thick. The requirements for manufacturing the surfaces relative to the datums were extremely tight. We first manufactured the concave surface resulting in a surface figure error relative to the datums of 23.6 μm (FIGURE 6). The best fit error with the major tilt error removed was smaller. This tilt error was determined to be too large based on the optical system requirements. To correct this error, we decided to correct machine the datums instead of correcting the surface figure error which would have taken too much time. The datum corrections will be done by machining the part's edges with a grinding process but the amount of material removed will be very small, less than 100 μm from the part's edge.

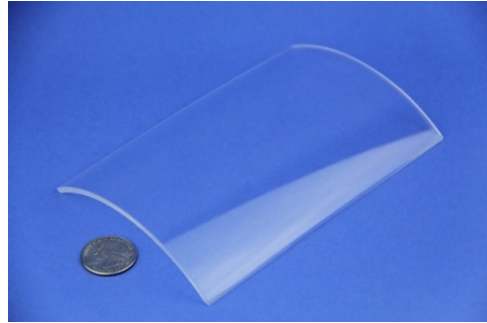


FIGURE 5: Example polished toroid optic, 102 mm x 152 mm, approximately 3 mm thick.

1. Start Condition

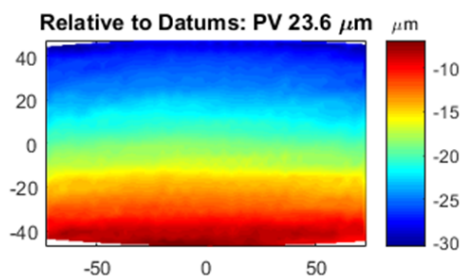


FIGURE 6: Start condition-small surface figure error, 94 mm by 144 mm.

Our original plan was to correct the tilt by machining the minus Y part edge. But, in performing that correction we determined that the part edges were not square (90°) to each other. This had to be corrected first, which resulted in a surface figure error of 14.8 mm shown in FIGURE 7. As shown, there now is twist (the astigmatism look) which we then decided to correct by rotating the x-y edge datums (the rotation was done on all edges to keep the datums square). Simulation showed that we need to rotate the edges by 0.012° to achieve minimal twist error. The result of this machining correction is shown in FIGURE 8. As indicated, the twist error *increased* to 22.6 μm . As the twist is in the same direction, it indicated that we rotated the datums in the wrong direction. By rotating the datums with machining the edges by 0.024° (the original error plus the wrong rotation induced error), the twist is removed as shown in FIGURE 9 with only 10.6 μm of error relative to the datums.

2. Machine Edge Square Up

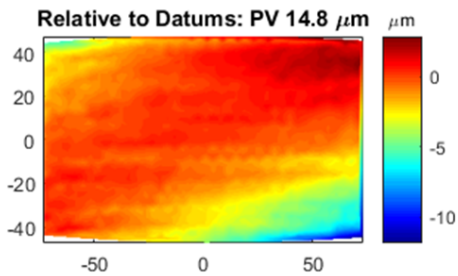


FIGURE 7: Condition 2 – squaring up the optic edges after polishing induced some twist in optic, 94 mm by 144 mm.

3. Machined edges to a rotation by $\sim 0.012^\circ$ CCW

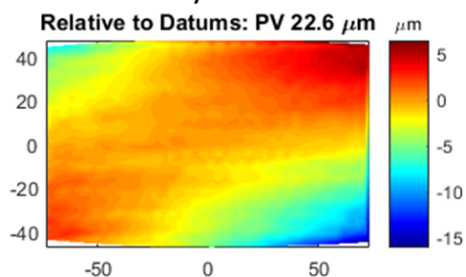


FIGURE 8: Condition 3 – Machining the edges with a new rotation, 0.012° CCW, 94 mm by 144 mm. Surface figure error gets worse, rotated the wrong way.

4. Machined edge datums by 0.024° CW.

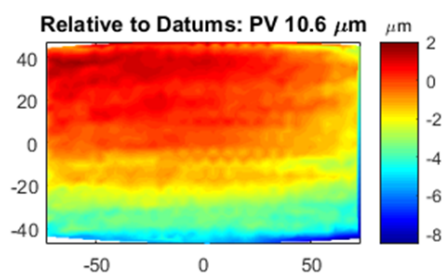


FIGURE 9: Condition 4 – Machining the edges with a new rotation, 0.024° CW (2 times the original error), 94 mm by 144 mm. Surface figure error improves, very little twist remains.

In conclusion, we have shown that it is feasible to correct the surface figure error (relative to the datums) by correcting the datums themselves.

Another feature that the fiducials permits is determining the surface to surface error.

FIGURE 10 shows the convex (top) and concave (bottom) surface deviations plotted illustrating the total deviation between the two surfaces. The concave data has been adjusted such that a deviation into the surface (less material than nominal) shows up as red. The color scale on the convex surface is such that extra material above nominal is red. In this way, we can visualize the deviation of each surface in the same direction. Red areas on both surfaces would indicate a global shift of both surfaces in the same direction. FIGURE 11 shows a plot of the surface normal thickness variation. The total thickness variation across the surface is $13.2 \mu\text{m}$ with an average deviation of approximately $4 \mu\text{m}$ from nominal. We attribute this success to the time spent shifting datums into the most optimum location before processing the second side. The concave surface is shifted down by 10 mm in the figure for ease of visualization.

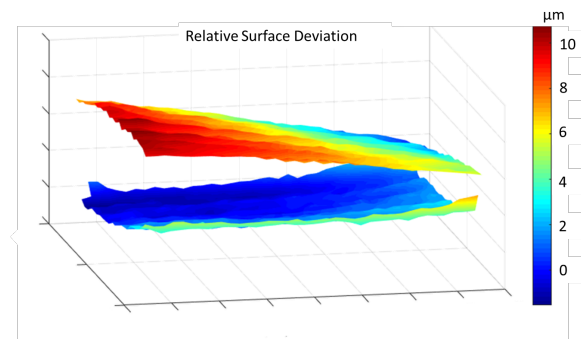


FIGURE 10: Relative surface deviation of the convex (top) and concave (bottom) 94 mm by 144 mm.

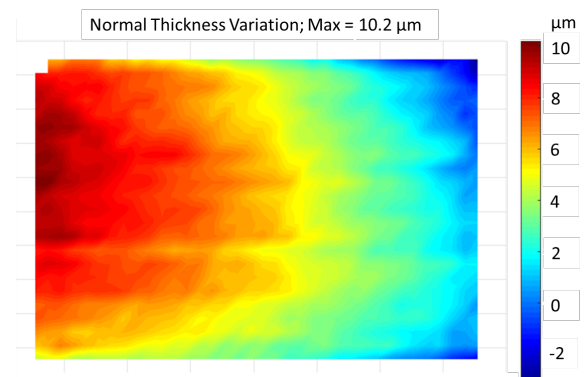


FIGURE 11: Thickness variation of the toroid like optic in FIGURE 5 Relative surface deviation of the convex (top) and concave (bottom) 94 mm by 144 mm.

Robot Accuracy Concerns

We have based much of our freeform polishing on a standard industrial robot, but questions

remain if the robot is accurate enough to polish precision optical components. Some initial polishing tests showed that we were accurate enough. Some positioning tests with a displacement interferometer (DMI), summarized here show that the robot is fairly accurate. Our goal for the DMI test was not to perform a full machine tool error matrix measurement, but rather to understand the expected errors during a typical polishing run. The DMI setup is shown in FIGURE 12. We measured only along a linear path, but as even a linear path requires movement by almost all robot joints we expect full volumetric errors to be of similar magnitude.

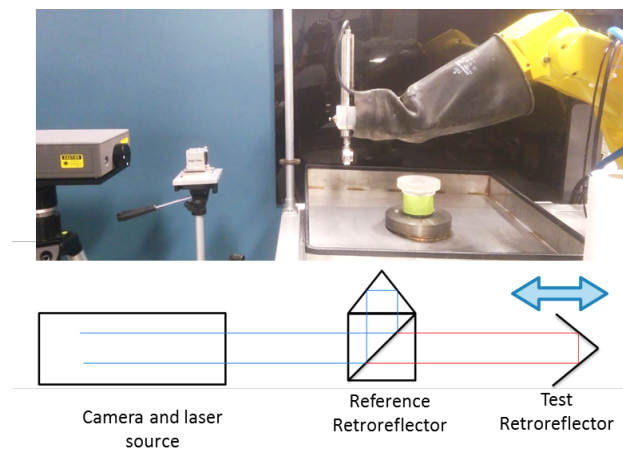


FIGURE 12: Displacement measuring interferometer setup on the robotic polishing platform.

We looked at both positioning and velocity errors during a typical tool path using two types of tests: steps and scans. In both the step and scan tests we varied the programmed step size. The step size is shown in FIGURE 13. The step size is the distance between the programmed way points along the tool path. This number is variable and is a choice that the operator can make when making a tool path. In the step test, shown in FIGURE 14 left, we programmed the robot to move to along a line in varying step sizes (each stair shows the robot moving) from ± 5 mm to ± 0.1 mm. Each stair step position was then held to look at the robot's ability to hold a set position and then the next position was moved to with the new step size. In the scan tests (FIGURE 14 right), we moved to a position 120 mm away using the varying steps sizes from ± 120 mm to ± 0.5 μ m. This test was used to evaluate positioning accuracy and speed accuracy.

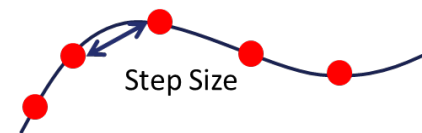


FIGURE 13: Definition of a step size when programming the robot – the distance between programmed way points.

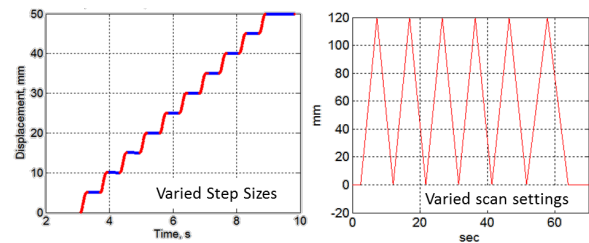


FIGURE 14: Programmed tests run: steps test – left: the +5 mm test is shown and right: the full scans test moving from 0 mm to 120 mm with vary step sizes (no pause on the steps).

FIGURE 15 shows the result of the step test where we programmed in a change in position using varying step sizes (distance between the way points). The data point shows the mean value of the error in the step (hundreds of steps were made). The error bars show a standard deviation in the step values. As shown and expected the errors and standard deviations are larger with the larger steps. This leads to a recommendation that smaller programmed step sizes gives better results, about ± 5 μ m. The bottom of FIGURE 15 shows the mean value and standard deviation in holding the step position (blue area in FIGURE 14). Again, the error and standard deviations are larger will larger step sizes, leading to a conclusion that there is a momentum effect. Overall, if we program with small step sizes we see about ± 5 μ m error in going to a set position and ± 3 μ m error in holding a set position. This is initial approximation of our probing accuracy (without doing a full uncertainty analysis).

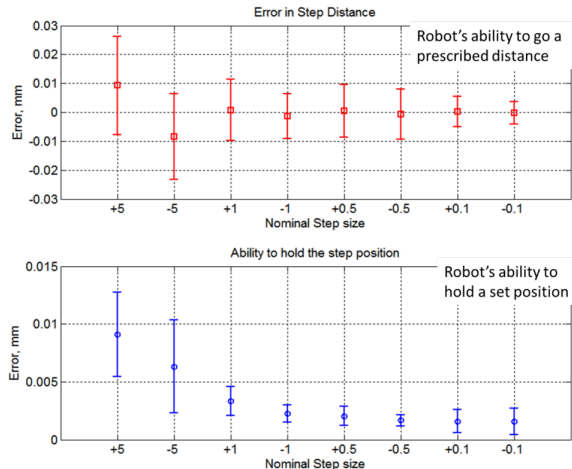


FIGURE 15: Results of step test. Top: the robot's ability to a prescribed distance. Bottom: robots ability to hold a set position. Nominal step size in mm.

The next test was the scan test - moving 120 mm with varying step sizes (distance between the way points). This test more approximates an actual tool path as there is no pause between the way points. The results of this test are shown in FIGURE 16. The top graph shows the actual position traveled, not the programmed position, but any error is buried in the linear path. The bottom graph shows the deviation for the linear path of three of the scans (at +24 mm, +1 mm, and -0.5 mm steps). As expected, the error is large for the large step sizes. The robot can follow a prescribed path better with smaller programmed way point distances. The error for step sizes of -0.5 mm is about $\pm 45 \mu\text{m}$ and +1 mm steps isn't much larger.

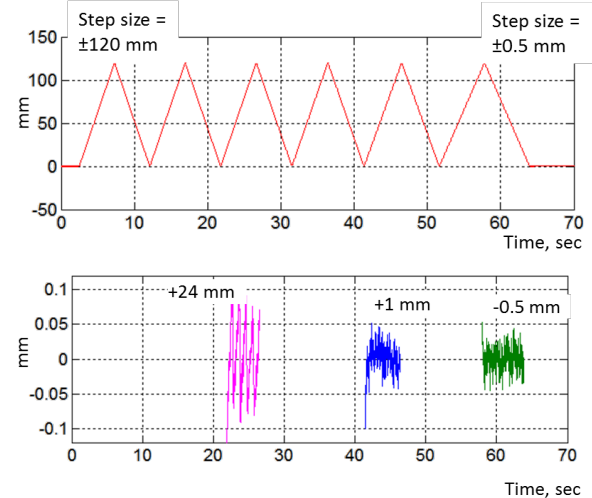


FIGURE 16: Results of scan test, top: actual positioned moved, bottom: linear component removed showing error in the programmed movement. Only three programmed step sizes are shown for clarity.

During the scan test, the robot should move at a speed of 25 mm/s. The actual measured instantaneous speed during the test is shown in FIGURE 17. This is for the same step size as in FIGURE 16. As shown, the mean speed drops for the smaller step sizes showing that the robot is not reaching up to full speed during the tool path. The speed variation is about 1.2 mm/s standard deviation, no matter the programmed step size.

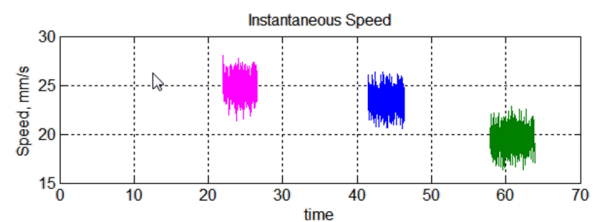


FIGURE 17: Instantaneous speed during the scan test.

The positioning and velocity tests had led us to the following conclusions:

- Polishing and probing should be done with 1 mm or 0.5 mm steps.
- Positioning accuracy is about $\pm 5 \mu\text{m}$
- The robot has the ability to hold a set position to $< 3 \mu\text{m}$
- Polishing velocity will be less than programed, but will be constant.
- We should not expect to correct surface errors that would correspond to changes in velocity of smaller than 1.2 mm/s.

These conclusions say that our robot positions better than the given specifications. We believe that the positioning accuracy along the path, about $\pm 45 \mu\text{m}$ is accurate enough for sub-aperture figure correction polishing. It is important to note that the figure correction algorithm depends on time, not on positioning. Rough positioning accuracy is required, but with the compliant tooling, the difference in positioning of $\pm 50 \mu\text{m}$ may not be too large an effect. Further tests would be interesting, possibility leading to correcting the positioning errors.

MULTIPLE FREEFORM SURFACE OPTICAL ELEMENT

An example freeform component that we have manufactured is shown in FIGURE 18. This optic is a monolithic telescope, meaning that it has multiple surfaces to image a target. Surfaces 3 and 2 are both freeforms, described as polynomial equations. Because this optic is monolithic, we do not have the ability to move the optical surface relative to each other during the assembly process. This optic is therefore a great example of why we require knowing the exact location of the optical surface in space relative to some datums features.

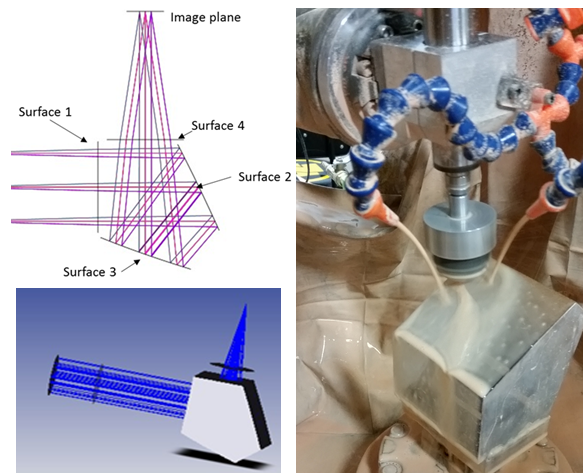


FIGURE 18: Example optic with multiple freeform surfaces.

STILL UPCOMING – MEASUREMENTS ON THE ROBOT

We are currently working on implementing deflectometry (fringe reflection) surface measurements on the robot. This will allow us to measure optics in-situ and quicker than the

CMM. The measurement system is not based on the robot's coordinate frame and therefore would not be subject to its errors. Initial tests show that image distortion calibration is a critical component in reducing the error in the measurement.



FIGURE 19: Fringe reflection setup on the robot polishing platform.

HIGH THROUGHPUT, HIGH RESOLUTION OPTICAL METROLOGY FOR NANOFABRICATED WIRE-GRID POLARIZERS

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ABSTRACT

A prototype imaging spectrophotometer system is discussed for use as metrology for high throughput roll-to-roll (R2R) nanophotonic device manufacturing, specifically nanofabricated wire-grid polarizers (WGP). The system characterizes 16 x 19 mm areas at a resolution of 8 x 8 μm enabling identification of defects smaller than individual liquid crystal display (LCD) pixels and subtle area-to-area variations that occur over device-scales. The throughput of the system is shown to be large enough to characterize 25% of an 80 mm wide roll of 10^5 contrast ratio (CR) WGP's moving at 0.1 m/minute. The combination of high throughput and high resolution allows for real-time detection of a variety of defects with unique signatures, enabling real-time root-cause analysis that identifies problematic fabrication steps paving the way towards advanced yield management for nanophotonic device manufacturing. The current throughput, current resolution, and the fact that the system can be further parallelized with multiple cameras suggests that this metrology can operate on rolls moving at multiple m/min and thus has suitable performance specifications to be used as metrology for nanophotonic film manufacturing for LCDs.

INTRODUCTION

Recent advances in nanofabrication are paving the way towards manufacturing a host of large-area nanophotonic devices on roll-to-roll (R2R) manufacturing lines. These devices include ultra-high contrast ratio (CR) wire-grid polarizers (WGP) [1,2,3,4] which can reduce power consumption in liquid crystal displays (LCD) by up to 20%, transparent metal mesh grids which enable flexible touch screens and displays [5,6,7,8], and nanoscale patterned plasmonic color filters which can simplify fabrication of color filters for displays and cameras into a single device layer [9]. The NASCENT center at The University of Texas at Austin is creating a prototype R2R nanofabrication line for the manufacturing of

these types of nanophotonic devices, in particular flexible WGP's with CRs exceeding 10^5 .

WGP's are a type of polarizer based on nanoscale metallic lines. WGP's can have extremely high CRs [2,3,4] and because they reflect s-polarized light rather than absorb it (as traditional organic-film polarizers do) WGP's can recycle light in displays improving power efficiency. For this reason WGP's have the potential to replace the traditional organic-film polarizers currently used as bottom linear polarizers in LCD displays. However, in order to compete with the current organic-film technology, it must be shown that WGP's can be made with similar device performance. WGP's must have a CR of 10,000 and a p-polarization transmittance (T) >85% at all visible wavelengths to compete in the current industry. Additionally it must be shown that WGP's can be manufactured on large-scale and at high yield. Clearly an advanced fabrication line is necessary, but in order for yield to be maximized metrology is needed as well.

The NASCENT Center is creating a prototype manufacturing line to fabricate WGP's on 80 mm wide rolls at 0.1 m/min. This is just a prototype system; in the future R2R nanophotonics lines could have rolls as wide as television monitors and may move as fast as multiple meters per minute. Metrology will have to keep pace with this level of throughput or else it will bottleneck production. That being said, it isn't necessary for factories to inspect 100% of their outgoing product. Only a fraction is typically characterized and the quality of the entire outgoing product can be inferred. An appropriate fraction must be determined for every unique manufacturing scenario. On mature factory lines just 1% may be suitable since expected defects are well-characterized, but prototype lines will require more complete characterization, maybe as high as 25%. This means the metrology needs to characterize 20 cm^2/min on NASCENT's prototype line, and the characterization must be done at high-spatial resolution so that small

defects and subtle area-to-area variations can be identified.

Traditional critical dimension (CD) metrologies like SEM and scatterometry will have extreme trouble hitting this target throughput. SEM, for instance, can only measure CDs with a field-of-view on the order of the CD itself, and thus characterizing many square cm's of device area per minute is impossible. Scatterometry can be used to characterize areas as large as 1 cm², but the measurement is a single average of the entire area, meaning that small defects and subtle area-to-area variations are nearly impossible to identify over large areas. In order for scatterometry to match the spatial resolution of the imaging spectrophotometer it would need to make millions of small-area measurements one-after-the-other. This would severely reduce its throughput, but high spatial resolution over large areas at high speed is necessary. CD metrologies cannot scale with R2R nanophotonics manufacturing.

Fortunately there is no inherent need to know the CDs of a nanophotonic device to be sure that it was made correctly. One only needs to test its performance using some sort of "functional" metrology. Functional metrology is routinely used in the semiconductor industry in the form of electrical testing of chips. An electrical test can be done in seconds which can qualify a chip comprised of billions of individual transistors, vias, and interconnects all with important CDs. One could argue that if a chip passes electrical testing it can be assumed to have been made correctly without needing to know the nanoscale CDs of the elements in the device. Now, because chip manufacturing lines have hundreds of individual fabrication steps CD metrology at intermediate steps is still necessary, but in nanophotonics manufacturing completed devices can require just a handful of fabrication steps (~5). Intermediate CD metrology steps will ultimately be more costly to the factory than a routine number of failed devices, because they would bottleneck throughput.

Functional metrology for nanophotonic devices can be done using spectrophotometry. Today spectrophotometers are ubiquitous [10,11,12,13], but traditional systems are inherently low throughput because they are based on single sensors, which limits these systems to measuring only one spot on a device at a time. This spot can be made quite large (multiple cm²), but similar to the case of

scatterometry, large spot sizes suffer from low resolution and are not capable of identifying point defects and subtle area-to-area variations. Fortunately the general principle of spectrophotometry is extendable to high resolution, highly parallel measurements by replacing the single sensor with a megapixel camera sensor. This is what is called imaging spectrophotometry and it allows for measuring millions of localized points on a nanophotonic device simultaneously. This creates a tremendous boost in throughput (millions of times) and makes this metrology ideal for R2R nanophotonics manufacturing.

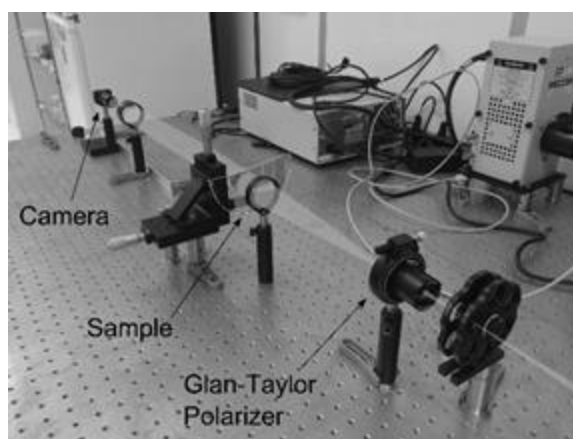


FIGURE 1. Image of Prototype Imaging Spectrophotometer.

RESULTS AND DISCUSSION

The System

A tunable light source from Newport [14] produces a collimated beam of monochromatic light with a minimum bandwidth of 0.7 nm at wavelengths ranging from 200-2400 nm. The beam is resized using a telescope lens assembly and is polarized using a Glan-Taylor reference polarizer. The beam is directed through a sample and the image of the light passing through the sample is focused onto a camera sensor using an objective lens. The camera is a 5 Megapixel CMOS from Point Grey [15]. The current setup allows for simultaneous characterization of 5 million localized areas of size ~8 x 8 μ m in a ~16 x 19 mm area of the sample.

Data Collection and Processing

Both the T and the CR of the WGP must be measured for full characterization. This requires three separate series of images: 1. Light transmitting through air (reference),

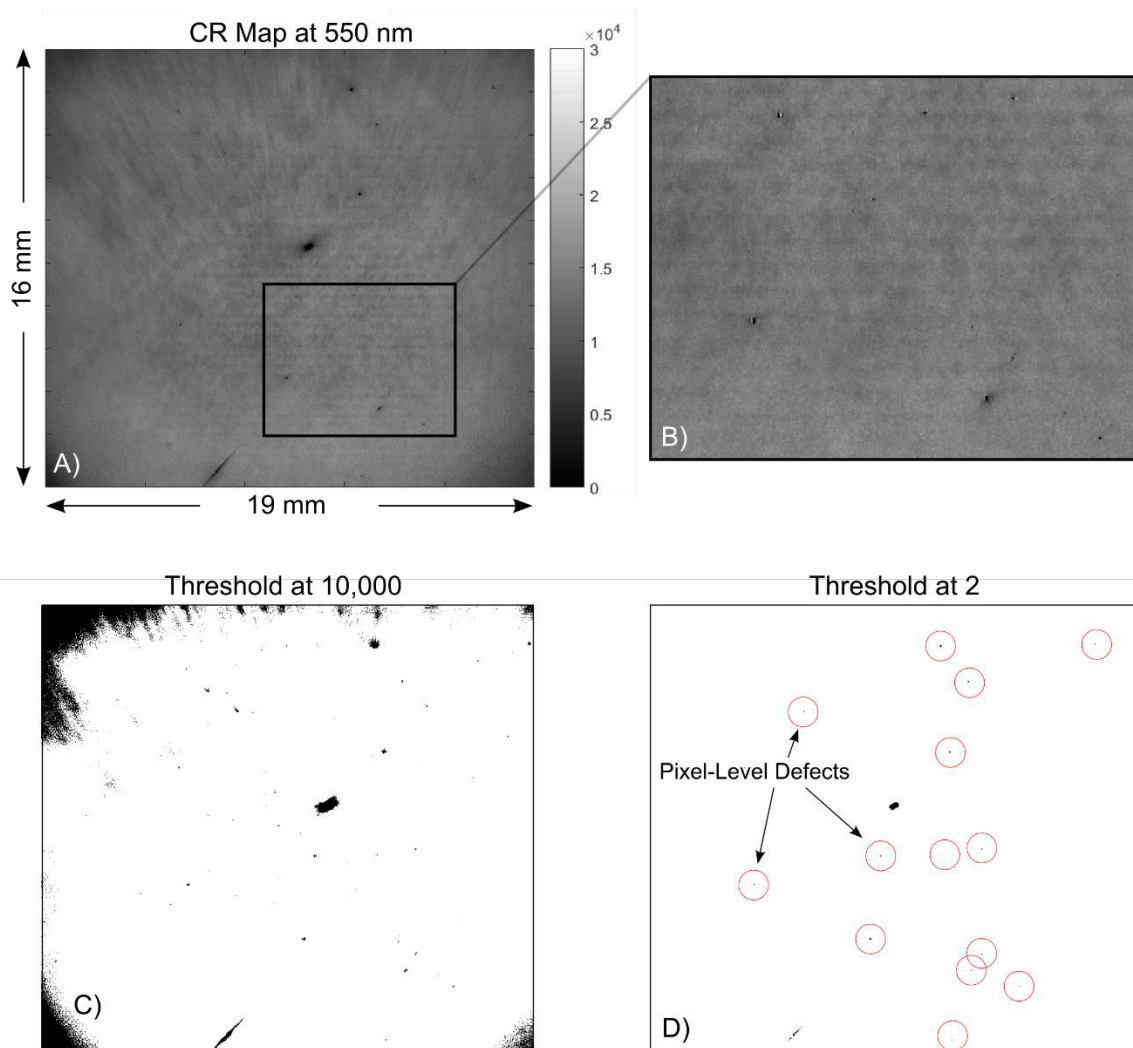


FIGURE 2. A) CR map of WGP at 550 nm. B) Close up which shows a variety of defects including point defects, horizontal streaks, and splotchiness. C) CR map thresholded at 10,000. Black areas are below threshold. Clearly shows that certain areas of the sample are not passing a typical inspection. D) CR map thresholded at 2. Scratches and pinholes as small as pixel-level (circled) are present.

2. S-polarized light transmitting through the sample (crossed state), and 3. P-polarized light transmitting through the sample (aligned state). All these images are taken at each wavelength of interest. The polarization of the incoming beam is changed by rotating the Glan-Taylor reference polarizer. In each of these image acquisitions, the exposure time used by the camera has to be set so that the camera “sees” the light transmitting through the sample, but doesn’t get “blinded” by it. An auto-exposure algorithm determines an appropriate exposure time for each individual measurement. In order for the measurements to be directly comparable, every pixel value is

scaled by the inverse of the exposure time used which produces a matrix of “intensities”. The T is the quotient of the aligned state and reference intensities. The CR is the quotient of the aligned state and crossed state intensities. These values are computed for every pixel producing a T and CR map at each wavelength of interest. An example of the CR map is shown in Figure 2A.

As shown in Figure 2C and 2D, the CR maps can be thresholded at various values to illustrate different types of metrology analysis including quality control (where the threshold is set at some inspection value) and defect identification (where

the threshold is set very low). Defects just a few camera pixels across are shown. Considering that the pixels in this image correspond to sample areas of $\sim 8 \times 8 \mu\text{m}$ this implies that defects much smaller than individual LCD pixels ($\sim 100 \mu\text{m}$) can be identified. Using a microscope objective the spatial resolution can potentially be extended all the way to the resolution limit of light, but significant sacrifices must be made on the total area of the measurement. It is possible that using a combination of measurement areas with different resolutions could be a helpful strategy for effective metrology. The current prototype system is a good balance between large-area measurements and high spatial resolution for our R2R system keeping in mind the end application of LCDs.

Throughput Analysis

There are many different sources of time affecting the throughput of this metrology. The majority of time is spent measuring the crossed state, because very little light transmits through the sample in this state, and in order for the camera to see this light it must use a very long exposure time. Based on previous measurements, a 10^5 CR WGP will require an exposure time of ~ 2.5 s in the crossed state. The aligned state however, transmits so much light that the camera can see it with an exposure time of less than 0.001 s. The reference measurement takes even less time than that. Some additional time is taken reading the image from the camera to the computer which is mostly dependent on the camera's frame rate; in this case 35 fps or 0.029 s per read. Each of these measurements must be done for every wavelength in the scan, and switching between wavelengths takes an additional ~ 0.1 s per switch. For each area of the sample being characterized the beam polarization must be changed once and moving between areas takes time as well. Currently the polarization change and sample movements are done manually and add about 30 s of time to the full characterization. Once automated however, these steps should be accomplished in ~ 0.5 s each. Thus the throughput of the metrology depends primarily on the CR of the WGP, the number of areas to be characterized, and the number of wavelengths in the scan.

There is an incentive to keep the number of wavelengths in the scan to a minimum because the long exposure done for the crossed state must be done at each wavelength. Fortunately,

characterization of WGP's may only require a small handful of wavelengths. Figure 4 shows the mean of the CR maps at wavelengths across the visible spectrum. The CR generally increases for longer wavelengths. This is because long wavelengths are less likely to leak through the WGP in the crossed state. If the qualification procedure simply requires that the CR be above a certain value at all wavelengths, it may be possible that WGP's only need to pass inspection at the shortest wavelength they will be operating at. However, it may be important for the CR at longer wavelengths to be much larger, in which case the WGP could be inspected at multiple wavelengths. Either way, it seems that a small handful of wavelengths is enough to fully characterize a WGP. Characterizing two wavelengths, for instance, would have a throughput of $19.6 \text{ cm}^2/\text{min}$ making it possible to characterize $\sim 25\%$ of NASCENT's R2R line. Extensions to larger, faster rolls in the future seem promising as well considering that smaller outgoing fractions will need characterization on mature factory lines. Furthermore, the imaging spectrophotometer can be parallelized by using multiple cameras which would improve throughput dramatically.

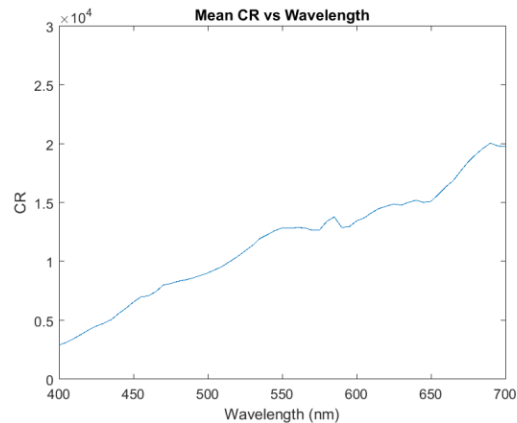


FIGURE 3. Plot of mean CR across the visible spectrum at 5 nm increments.

FUTURE WORK

The maps like those in Figure 2 contain a wealth of information including subtle area-to-area variations, scratches, and point defects. Identification of these different types of defects can provide crucial information to help yield management. For instance the subtle CR variations that look like horizontal streaks seen best in Figure 2B suggest residual layer thickness

TABLE 1. Different Types of Defects, Their Signatures, and Root Cause.

Defect Type	Spatial Signature	Measurement Signature	Test	Root Cause
Point defect	Localized, Circular, Variety of scales	CR $\rightarrow$ 1, T $\rightarrow$ 100%	Threshold CR at 1, Threshold T at 100%	Particle, template defect, or scratched
Scratch	Localized, Linear, Variety of scales	CR $\rightarrow$ 1, T $\rightarrow$ 100%	Threshold CR at 1, Variety of scales	Particle, template defect, or scratched
Linewidth variation	Usually mm scale	Subtle CR and T variations around expected values	CR and T delta	RLT variation or template feature variations
Lineheight variation	Usually wafer scale with significant edge effects	Subtle CR variations around expected value, T unaffected	CR and T delta	Etch rate variation

(RLT) variations in the nanoimprint process, because the symmetry and periodicity of these variations is consistent with known RLT variations. Image processing of the CR and T maps can be used to identify patterns which signify specific types of defects. These defects can oftentimes be linked to specific fabrication steps and thus this data can be used to perform root-cause analysis in real-time on the manufacturing line. This will be of critical importance in yield management on R2R nanophotonics lines. A variety of different defects are summarized in Table 1.

CONCLUSION

A prototype imaging spectrophotometer system has been discussed for use as metrology for high throughput R2R nanophotonic device manufacturing, specifically nanofabricated WGs. The system characterizes 16 x 19 mm areas at a resolution of 8 x 8 μ m enabling identification of defects smaller than individual LCD pixels and subtle area-to-area variations that occur over device-scales. The throughput of the system has been shown to be large enough to characterize 25% of an 80 mm wide roll of 10⁵ CR WGs moving at 0.1 m/minute. The combination of high throughput and high resolution allows for real-time detection of a variety of defects with unique signatures, enabling real-time root-cause analysis that identifies problematic fabrication steps paving the way towards advanced yield management for nanophotonic device manufacturing. The current throughput, current resolution, and the fact that the system can be

further parallelized with multiple cameras suggests that this metrology can operate on rolls moving at multiple m/min and thus has suitable performance specifications to be used as metrology for nanophotonic film manufacturing for LCDs.

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Development of a Twyman-Green Interferometer Capable of Measuring the Thermal Properties of Optical Materials

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ABSTRACT

When the temperature of an optical element changes, both its physical dimensions as well as its index of refraction change as a result. When designing an optical system that operates across a broad range of temperatures, knowledge of these thermal properties is critical. An interferometric method is presented for simultaneously measuring both the coefficient of thermal expansion (CTE) and the temperature-dependent refractive index (dn/dT) of materials in air. Any instrument that characterizes temperature-dependent properties must itself be largely insensitive to those same temperature changes. The design and realization of such an interferometer is detailed, highlighting the methods used for mitigating vibrations and undesirable temperature gradients. A discussion of measurement results on a Mitutoyo steel gauge block and CaF_2 is also included.

Background

Interferometric techniques have been utilized previously to measure CTE and dn/dT over a range of temperatures and wavelengths [1-5].

The change in dimension of a sample due to a change in temperature can be represented by

$$L' = L(1 + \alpha \Delta T),$$

where α is the CTE, L' is the new length, and ΔT is the change in temperature. Measurement of these parameters produces a plot of the change in thickness as a function of temperature. The slope of this plot is the linear CTE.

A similar relationship exists for the determination of dn/dT . The refractive index of a sample can be expressed as

$$N'_{00} = N_{00} + \left(\frac{dn}{dT}\right) \Delta T,$$

where N_{00} is the reference temperature refractive index of a sample [6]. N'_{00} is the index of refraction of the sample after a temperature change of ΔT , and dn/dT is calculated in a similar manner to α .

Instrument Description

The intended measurand of this instrument is a material's thermal properties at a variety of wavelengths, from the visible ($0.458 \mu\text{m}$) into the infrared ($3.39 \mu\text{m}$), in the temperature range from -40°C to $+80^\circ\text{C}$. Measurements in this paper are reported at a wavelength of $0.6328 \mu\text{m}$. One noteworthy characteristic of this apparatus is that material properties are measured in air.

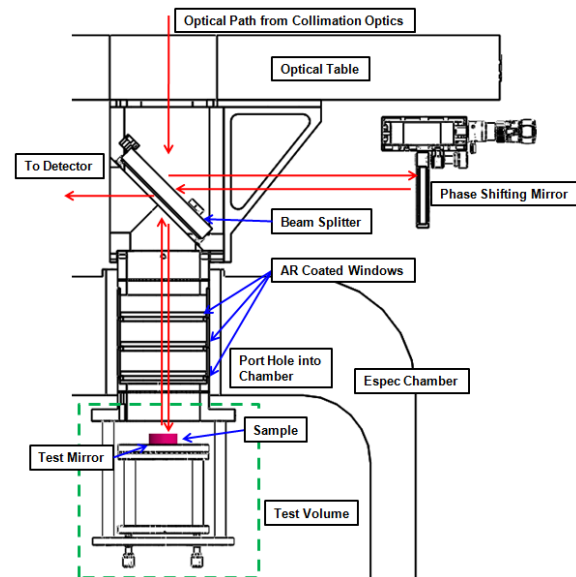


FIGURE 1. Schematic of the interferometer. The elements outlined in green are inside a chamber that controls temperature and humidity.

The system developed presently consists of a Twyman-Green interferometer in which the sample is placed inside an Espec BTX-475

environmental chamber as shown in Figure 1. This chamber provides a test volume for varying the temperature of the sample. The CTE and dn/dT are simultaneously measured by tracking the optical path differences induced by the material's response to temperature.

The optics on top of the optical breadboard deliver a 50 mm diameter collimated laser beam vertically downward to the plate beamsplitter. The light that is reflected by the beamsplitter is directed toward the reference mirror. The reference mirror is attached to a Thorlabs Nanoflex stage, with a piezo-electric driving element run in an open-loop configuration. This component allows phase-shifting to be performed on the interferometer, which enables the tracking of optical path changes over time. For phase-shifting a 25-step least-squares algorithm is used, based on a technique by Malacara [7].

The light that is transmitted by the beamsplitter travels downward through a port into the environmental chamber. This ensures that the only components of the instrument exposed to temperature gradients are the sample and its supporting mechanical test arm. The test arm contains, along its length, anti-reflection-coated windows which further thermally isolate the environmental chamber from the optical components above. The rest of the interferometer remains at ambient temperature throughout the duration of the test. The optical components of the interferometer are mechanically decoupled from the Espec chamber via four passive isolation feet attached to an aluminum frame.

Photographs of the interferometer are shown in Figure 2. The photograph in the left of Figure 2 shows the aluminum frame which supports the optical breadboard. The photograph in the right of Figure 2 shows the thermal circuit that exists as part of the test arm of the interferometer. The significance of this will be discussed in greater detail below.

Figure 3 shows the three separate beam paths under consideration during the measurement. The difference in optical path between the reference mirror and test mirror is OPD_1 , referred to as the "background". The difference in optical path between the reference mirror and the top of the sample is OPD_2 . This region of the sample possesses a 50-60 nm thick gold

reflective coating, making it possible to measure CTE. This coating is negligibly thin compared to the sample under test. The difference in optical path between the reference mirror and the test mirror, where the beam passes through the transmissive part of the sample, is OPD_3 . This region enables the measurement of dn/dT . The background measurement allows for a reliable comparison between the optical paths in spite of a noisy environment.

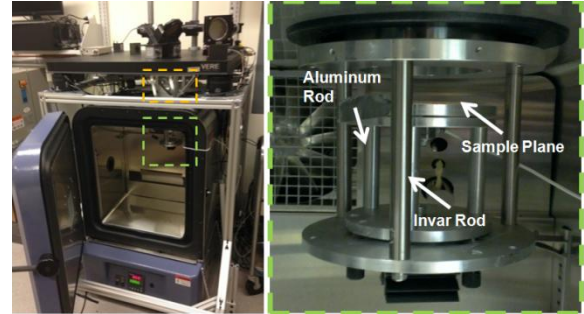


FIGURE 2. Photographs of the as-built instrument. Left: Thermal Interferometer system. The green box shows the test volume, and the orange box shows the beamsplitter. Right: Enlarged view of test volume.

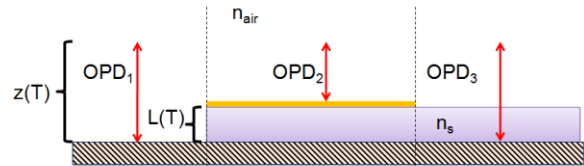


FIGURE 3. Diagram of a transmissive sample under test, resting upon the interferometer's test mirror. The three different beam paths are used to extract the material properties of the sample.

The mathematical expressions for the lengths of the three optical path differences are

$$OPD_1(T) = 2 \cdot n_{air}(T) \cdot z(T),$$

$$OPD_2(T) = 2 \cdot n_{air}(T) \cdot [z(T) - L(T)],$$

$$OPD_3(T) = 2 \cdot n_{air}(T) \cdot [z(T) - L(T)] + 2 \cdot n_s(T) \cdot L(T).$$

where $n_{air}(T)$ is the refractive index of air, $n_s(T)$ is the refractive index of the sample, $z(T)$ is the optical path difference between the reference and the test mirror, and $L(T)$ is the length of the sample. It is emphasized that each of these quantities is a function of temperature.

The air temperature and the temperature of the test mirror are measured with K-type thermocouples. The temperature of the sample is assumed to be the temperature of the test mirror. For samples less than 10 mm thick, this assumption holds to within the resolution of the thermocouples (0.1°C). Direct measurement of the sample's temperature is undesirable due to the lateral forces exerted by the thermocouple wire, along with the potential for contamination of the sample by adhesives and thermally conductive paste.

One concern addressed by this instrument is the optical path change due to the test arm's thermal expansion, and also from the refractive index change of air along the test arm. To illustrate the magnitude of this latter effect, Figure 4 shows the theoretical change in optical path from the change in air temperature alone, modeled using equations of the index of refraction of air as a function of temperature from NIST [8]. For comparison, the change in optical path caused by the thermal expansion of a steel block, 10 mm thick, is also plotted.

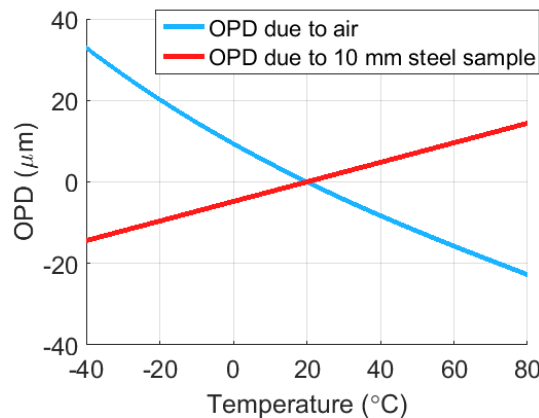


FIGURE 4. Optical path change due to air for a 200 mm cavity.

Since the instrument's sensitivity to temperature changes is on the same order as the sample's temperature sensitivity, the thermal compensating circuit seen in the right of Figure 2 is employed. The goal is to design the system so that the piston change of the background (OPD_1) is minimized. As explained in the detail of [9], the thermal expansion of the invar rods tends to displace the sample plane downwards. The aluminum mirror mount constitutes a negative compensating length, which tends to displace the sample plane upwards. This mirror mount sits on three stainless steel screws, which serve

two functions. First, the screws compose a kinematic interface, keeping the mirror centered on the optical axis during the experiment, while also providing tip and tilt adjustments of the test mirror. Second, the screws allow the thermal behavior of the instrument to be fine tuned by adjusting the length of material in the compensating part of the thermal circuit.

Theoretically, it is possible to minimize the instrument's sensitivity to temperature to less than $2\text{ }\mu\text{m}$ over the entire temperature range. The majority of this residual sensitivity is due to the nonlinear change in air's refractive index.

Test Procedure

Samples appropriate for test with this method are between 2-25 mm thick with two parallel, polished faces. The length L of the sample is measured with a pair of calipers. It is important that the amount of wedge on the samples is low, otherwise it will not be possible to resolve fringes on the test mirror and the sample at the same time.

The sample is placed on top of the test mirror so that the camera sees interference patterns such as the one in Figure 5.

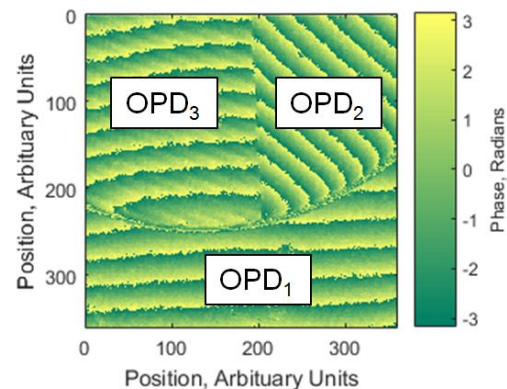


FIGURE 5. An example of the three interference patterns used to characterize a material. OPD_1 : Background (reflected from the test mirror). OPD_2 : Reflected from the top of the sample. OPD_3 : Transmitted through the sample.

The process for making a measurement is as follows: first, the chamber cools down to the lowest temperature during the test, over the course of three hours. Once the system has soaked at this setting long enough to reach equilibrium, logging of temperature and phase data begins. Temperature is polled every 5 seconds, while phase is recorded once per 0.8

seconds. The temperature is then increased at a rate of 0.07 °C/min up to the maximum test temperature, while the instrument is active.

An alternative test scheme can be adopted in which the test chamber is brought to the hot or cold temperature, and then turned off. The sample and chamber slowly return to ambient temperature in a quasi-steady state fashion. While the chamber is off, vibration effects are minimized.

Measurement Results

To understand the sources of error in the interferometer, data was collected for an 18 hour period with no sample present in the test volume. The measured optical path difference for this test is shown in Figure 6.

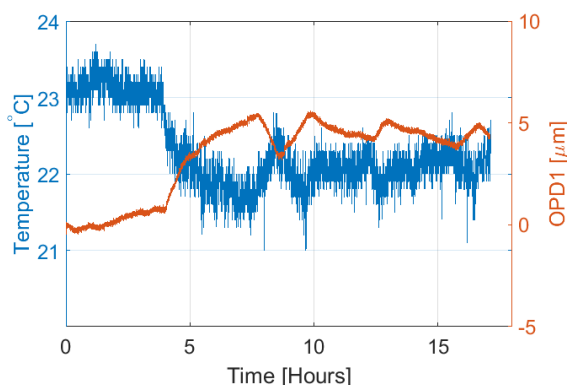


FIGURE 6. The measured change in optical path within the interferometer's test arm over an 18 hour period, compared to the temperature measured in the interferometer's reference arm.

The measured signal in the background path (OPD_1) gives a value of 1.88 μm RMS for the drift of the instrument. Ambient temperature variations in the reference arm of the instrument account for this drift. A typical testing cycle over the entire temperature range occurs over a period of 12-14 hours. As discussed earlier, this drift is present across the entire aperture of the instrument, and so subtracting the background signal from OPD_2 and OPD_3 allows for correcting the measurements of CTE and dn/dT .

Figures 7 shows the measurement of CTE for a CaF_2 sample that is 5.28 mm thick [10]. The temperature range examined is -10°C to +50°C with plotted increments of 10°C. Measurements agree with those made by Corning and Cardinali [10,11,12].

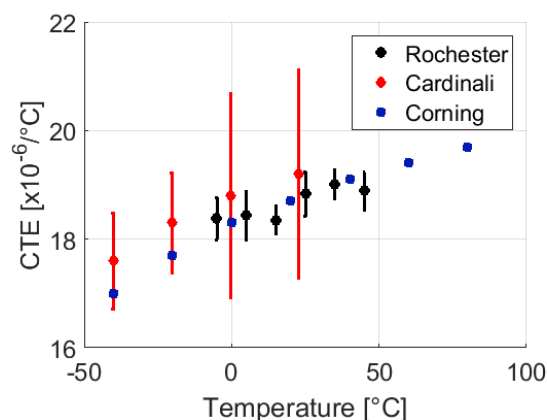


FIGURE 7. Measurement of the CTE of CaF_2 [10-12].

Figure 8 presents the results for the measurement of dn/dT in the CaF_2 sample. These measurements agree with the result by Cardinali, but there is a discrepancy between the data reported by NASA and Corning.

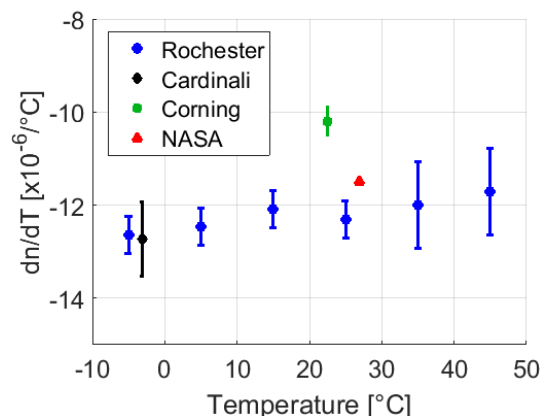


FIGURE 8. Measurement of the dn/dT of a 5.28 mm thick CaF_2 sample [10-13].

Figure 9 presents measurements of a 20 mm thick Mitutoyo steel gauge block. The gauge block is Grade 00 per ASME B89.1.9-2002 and is NIST-traceable. This measurement is compared to the manufacturer specification, and to measurements made by Okaji *et al.* [14]. The plots labeled S1, S2, S3, and S4 are for four separate 100 mm thick samples that were measured by Okaji. The variability that Okaji *et al.* report for this material from sample to sample is in good agreement with the error bars on the Rochester data. For comparison, the measured 20 mm gauge block is certified to possess a CTE of $10.8 \pm 0.5 \times 10^{-6}/^\circ\text{C}$ at 20°C. The measurements performed on this system are well within this specification.

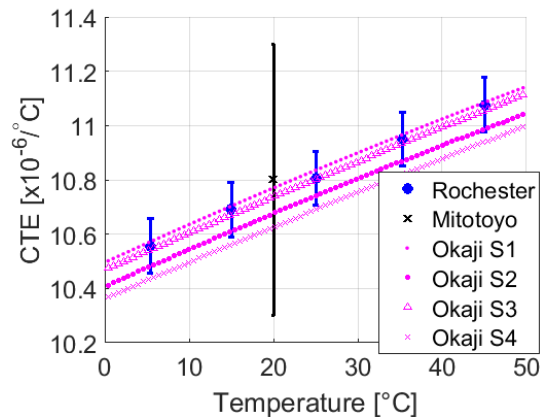


FIGURE 9. Measurement of the CTE of a Mitutoyo steel gauge block compared to values in literature [2].

Future Work

Presently, this interferometer is capable of measuring in the range from -40°C to $+50^{\circ}\text{C}$. It is also capable of making measurements at other wavelengths for this temperature range.

The temperature range of this instrument is limited on the cold end by the formation of condensation on the float glass windows in the test arm, and on the hot end by the turbulent mixing of air within the test chamber. At cold temperatures, mixing is not a problem. For temperatures above 50°C , the density of air in the test chamber is significantly lower than the laboratory air, and this causes it to escape out of the port hole in the top of the Espec chamber. Attempts to seal this gap couple vibrations into the interferometer, and as a result, this trade-off is a fundamental limit of the current design.

Development of the instrument is on-going and the first tests of a second prototype are currently under way. So far, the new prototype boasts a much lower noise floor, a more stable temperature control, and a larger available temperature range.

Studies of spatial uniformity have begun to quantify this method's readiness to measure a gradient index (GRIN) material. The qualifying test will consist of a Zerodur (nominal CTE of $0.0 \times 10^{-6}/^{\circ}\text{C}$) sample measured in different orientations, and the values of each test compared for spatial bias. Preliminary measurements have shown that Zerodur produces a very small distinction between OPD_1 and OPD_2 , making it an interesting topic of study with respect to the interferometer's noise floor.

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Experimental Validation of Complex Non-Minimum Phase Zeros in Flexure Mechanism Dynamics

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BACKGROUND AND OBJECTIVE

This research is motivated by the need to achieve large range, high precision, and high speed – all simultaneously – in multi-axis flexure mechanisms. Previous work has shown large range (10mm per axis) and nanometric precision (25nm) in an XY motion system based on a parallel-kinematic flexure mechanism [1]. This mechanism employs a systematic and symmetric layout of eight double parallelogram flexure modules (DPFM), resulting in a high degree of geometric decoupling between the X and Y motion axes. But experimental frequency response measurements of the X direction transfer function relevant for motion control exhibit complex non-minimum phase zeros (CNMP) at certain Y direction operating points. These CNMP zeros are highly detrimental to the closed-loop dynamic performance of the motion system including bandwidth and stability robustness [2].

To understand the conditions under which such CNMP zeros arise and determine if physical design decisions can eliminate these zeros, a representative XY flexure mechanism (Fig. 1) has been recently investigated [3]. This design is relatively simpler but has all the essential attributes of the original XY flexure mechanism [1], including multi-axis motion capability (X and Y) and at least two DPFMs in a nominally symmetric configuration that lead to closely spaced modes. The non-collocated transfer function from the force P to displacement X_1 was examined, as a function of cross-axis operating point Y_{1o} , via lumped-parameter non-linear dynamic modeling. A static Y direction force F_0 was assumed on the motion stage (stage 1) to establish this non-zero Y direction

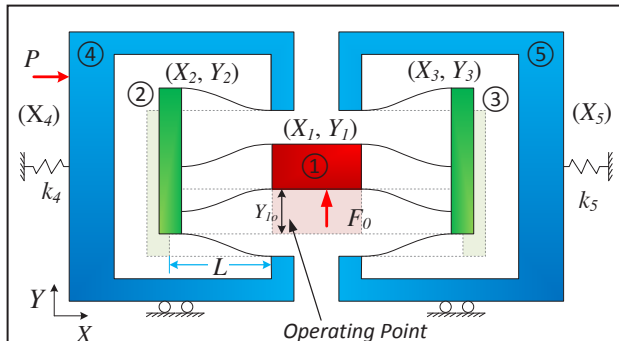


Fig.1. Simple Representative 4 Degree-of-Freedom (SR4DoF) XY Flexure Mechanism

operating point (Y_{1o}). It was shown that the geometric non-linearity associated with arc-length conservation in flexure beams along with the kinematic under-constraint in the DPFM result in a coupling between the Y motion of the secondary stages (2 and 3 in Fig.1) and the X motion of the motion stage for non-zero operating points (Y_{1o}). Furthermore, the two closely spaced modal frequencies associated with the Y motions of the secondary stages of the two DPFM were varied by introducing an intentional parametric asymmetry (mass difference between stages 2 and 3, given by $\Delta m_{23} = m_2/m_3 - 1$) to simulate practical manufacturing tolerances.

This model showed that these closely spaced modes interact as the above mentioned coupling varies with the operating point and as the parametric asymmetry varies, giving rise to CNMP zeros under certain conditions. This is illustrated in Fig.2, which maps the existence of CNMP zeros against a range of operating points (Y_{1o} normalized with respect to beam length L) and parametric asymmetry (Δm_{23}). Black regions on this map indicate the presence of CNMP zeros. This map indicates that one can intentionally choose the physical design parameters (e.g. $\Delta m_{23} < 0$) to eliminate the CNMP zeros, enabling better dynamic performance (i.e. higher speed, bandwidth, and stability robustness). This paper presents an experimental validation and confirmation of these modeling predictions.

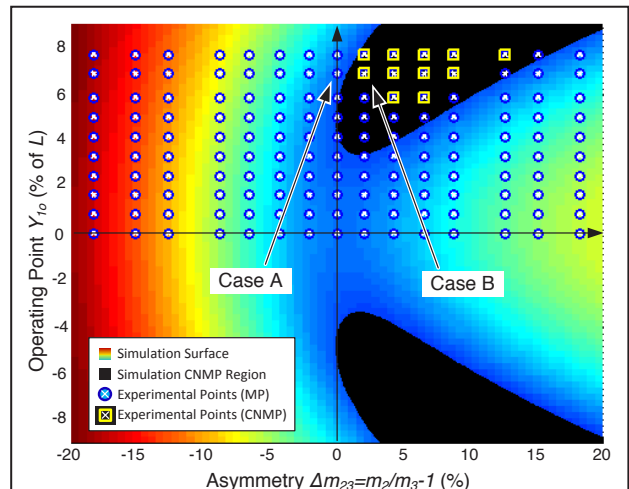


Fig.2. Complex Non-Minimum Phase Zero Map

EXPERIMENTAL SETUP DESIGN

The goal of the experimental setup design is to minimize factors such as noise, disturbance, undesired modes, parasitic dynamics, etc. so that one can focus on and validate the specific CNMP phenomenon predicted by the above referenced model. Key hardware design challenges addressed in this effort are: (i) The theoretical roller and spring functionalities in Fig.1 are achieved via simple flexure bearings to eliminate friction and backlash; (ii) The geometry and dimensions of the overall flexure mechanism are iteratively selected to isolate the modes associated with the secondary stages 2 and 3 from other modes. Isolating these two “modes of interest” helps provide a “clean” observation of their interaction and allows an investigation into the genesis of CNMP zeros; (iii) The constant Y direction force (F_0) is implemented via a long steel cable loaded by a free-hanging weight with a virtual pulley [4], which effectively prevents parasitic dynamics and offers frictionless loading.

Overall Flexure Mechanism Geometry

For ease of discussion, we use the term *main flexure* to refer to stage 1 through stage 5 in Fig.1, including the two DPFMs. Fig.3 illustrates the design evolution of the overall flexure mechanism, comprising the main flexure that is designed for conducting the relevant experiments. Starting with Fig.3a, the initial configuration of the SR4DoF flexure mechanism is shown using two springs and rollers that connect the main flexure to the ground. In order to eliminate friction and backlash, the springs and rollers are replaced by four simple flexure beams in Fig.3b. However, such design has limited travel range in the X direction due to geometric over-constraint of the flexure beams. To

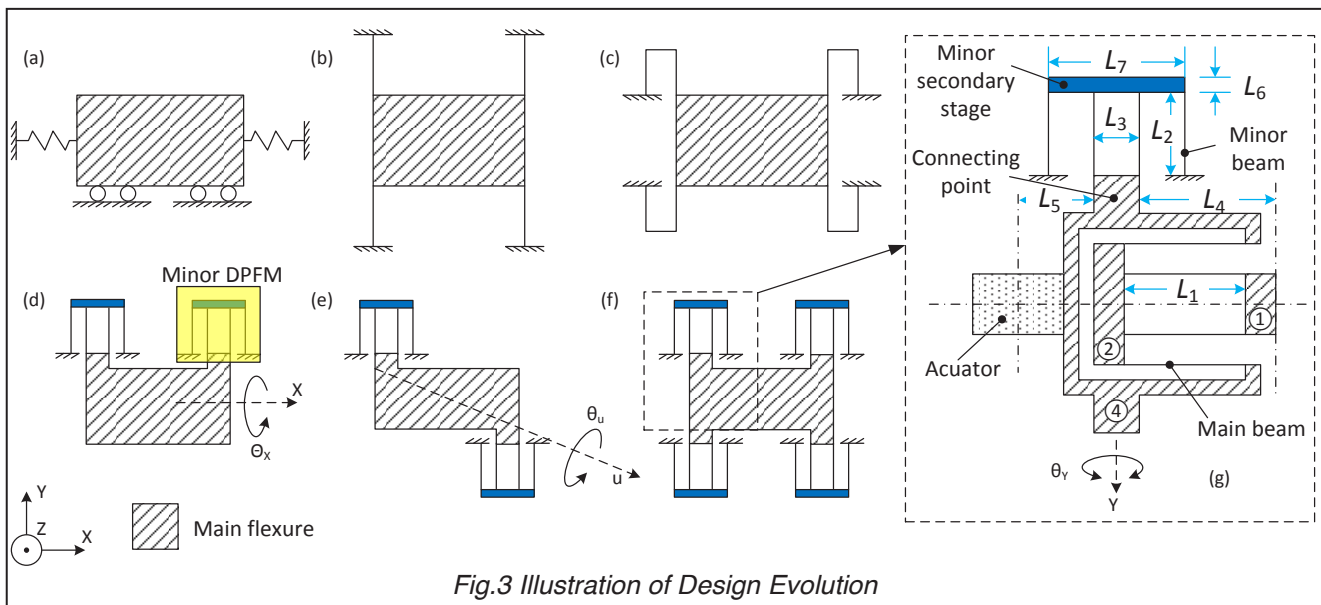
relieve this over-constraint, the simple beams are replaced by folded beams in Fig.3c. However, the folded beams offer low translational stiffness along the Y-axis and low rotational stiffness about the Z-axis. This leads to additional undesired low frequency modes. In Fig.3d, these folded beams are replaced with DPFMs, since the latter offer better stiffness characteristics without sacrificing travel range. In order to avoid rotational modes about the X axis, as shown in Fig.3d, or rotational modes rotation about the U-axis, as shown in Fig.3e, four *minor* DPFMs are used symmetrically in the design of Fig.3f. The term *minor* is used in order to distinguish these four DPFMs from the two DPFMs in the main flexure of Fig.1. The tradeoff here is a slightly higher stiffness (k_4 and k_5) in the X direction of the overall flexure mechanism, which leads to a higher frequency of the X direction rigid body mode of the overall flexure mechanism. This has to be taken into consideration as we aim to isolate the two modes of interest associated with the secondary stages of the main DPFMs from rest of the modes, as described next.

Overall Flexure Mechanism Dimensions

Once the overall flexure mechanism design was conceptually established, the detailed dimensions were optimally selected as follows.

(1) We started with choosing the dimensions of the main flexure beams (Fig.3g) to be the same as that in the original XY flexure mechanism that motivated this investigation [1, 3]. Accordingly, the length of the main beam was set at $L_1=47.5\text{mm}$ and thickness was set at $T_1=0.625\text{mm}$.

(2) The mass of each stage in the main flexure (Fig.1) was selected so as to isolate the two modes of interest (3rd and 4th modes in Fig.4) from the first two modes (1st and 2nd modes in Fig.4). This was



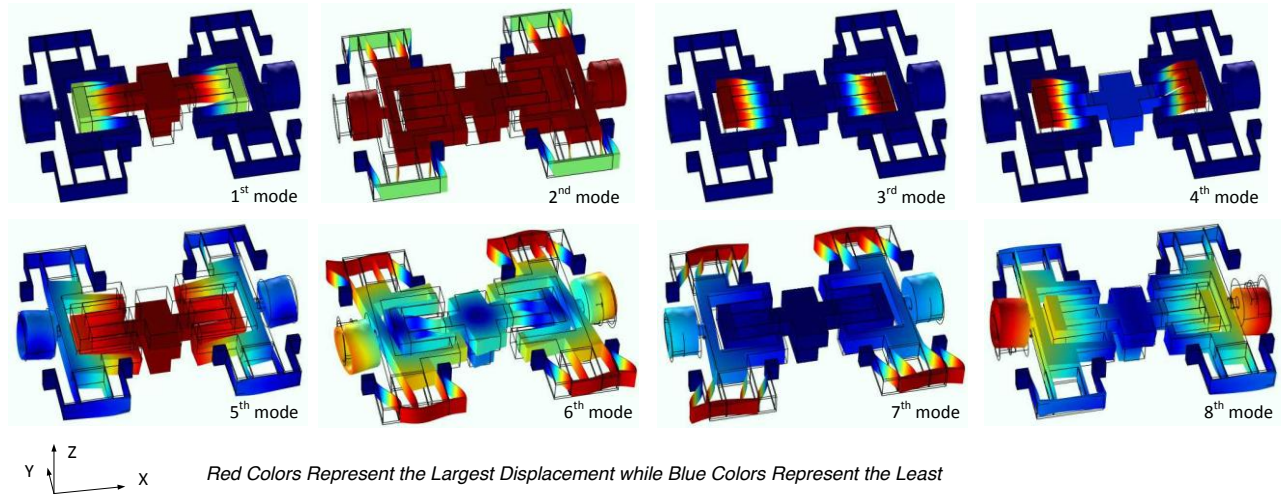


Fig.4 Mode Shapes of the Final Design

TABLE.1 Key Dimensions, Corresponding Natural Frequencies and Mode Orders

Dimensions (mm)						Mode order and natural frequencies (Hz)							
L ₂	T ₂	L ₃	L ₄	L ₆	L ₇	1 st	2 nd	3 rd	4 th	5 th	6 th	7 th	8 th
47.5	1	40	85	3	80	21	31	80	84	213	231	271	272

accomplished by setting $m_1=0.4\text{kg}$, $m_2=m_3=0.06\text{kg}$, and $m_4=m_5=0.78\text{kg}$. The 3rd and 4th modes are the closely spaced modes of interest that we intend to investigate in this study.

(3) Furthermore, it was desirable to set the natural frequencies of the modes associated with the vibration of the secondary stages of the four minor DPFMs to be much higher than the two modes of interest. To increase these resonance frequencies, one can either increase the transverse stiffness of the minor beams or reduce the mass of the minor secondary stages. The former approach increases the frequency of primary X direction rigid body mode (2nd mode in Fig.4) bringing it closer to the modes of interest, while the latter results in a smaller L_6 , which increases the compliance of the minor secondary stages and introduces additional undesired modes at lower frequencies. This tradeoff was addressed by iteratively selecting dimensions and checking the mode shapes and frequencies. In the final design, we chose $L_2=47.5\text{mm}$, $T_2=1\text{mm}$ (thickness of minor beam), $L_6=3\text{mm}$, and $L_7=80\text{mm}$.

(4) The choice of dimensional values for L_3 , L_4 and L_5 (see Fig.3g) is based on the consideration that the rotational modes about the Y and Z axes should be kept much higher than the modes of interest. Therefore, the span L_3 of the inner minor beams has to be large enough (within layout size constraints) to ensure that rotational stiffness of the minor DPFM about the Y and Z axes is high. Furthermore, dimensions L_4 and L_5 have to be chosen so that moment of inertia contributed by stages 1, 2, 3, 4, and 5 about the Y and Z axes is minimized. Accordingly, we chose $L_3=40\text{mm}$,

$L_4=85\text{mm}$ and $L_5=30\text{mm}$ in the final design to meet the requirements outlined here.

Summary of Mode Orders and Frequencies

Table.1 summarizes the dimensions selected for the final design, and the resulting mode orders and frequencies of the overall flexure mechanism (Fig.3f).

The 1st mode is the in-phase Y direction vibration of stages 1, 2 and 3. Therefore, this mode makes a negligible contribution to the frequency response measurement at stage 1 in the X direction. The 2nd mode is the primary X direction 'rigid-body' of the overall flexure mechanism. This mode shows up as the second mode due to the increased X direction stiffness of the minor beams, based on criterion (3) discussed above.

The 3rd and 4th modes are the modes associated with the vibration of two main secondary stages, which are the modes of interest in this paper.

The 5th mode is an out-of-plane mode in the Z direction while the 6th mode is an in-plane rotational mode about the Z direction. Next, there are four modes associated with the X direction vibration of the four minor secondary stages. These are the 7th (271 Hz), 9th (283 Hz), 12th (286 Hz), and 14th (307 Hz) modes. There are several other modes in between these four modes given the choice of dimensions. For example, the 8th mode comprises out-of-plane rotation about the Y direction.

In summary, in the final design, there is at least a two-times frequency separation before and after the closely spaced modes (3rd and 4th) of interest,

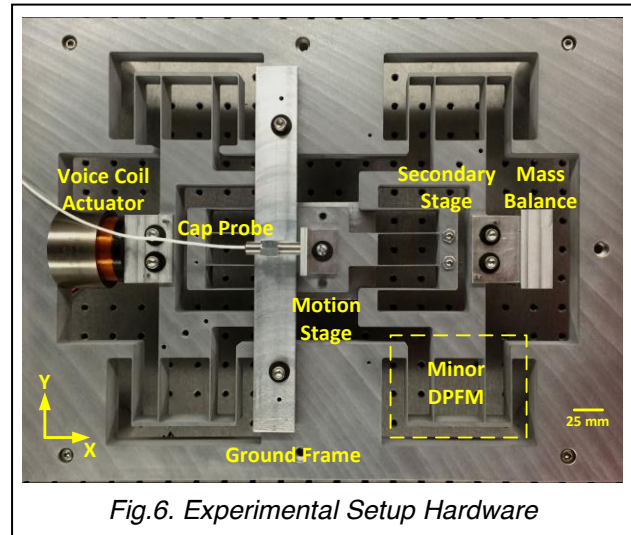
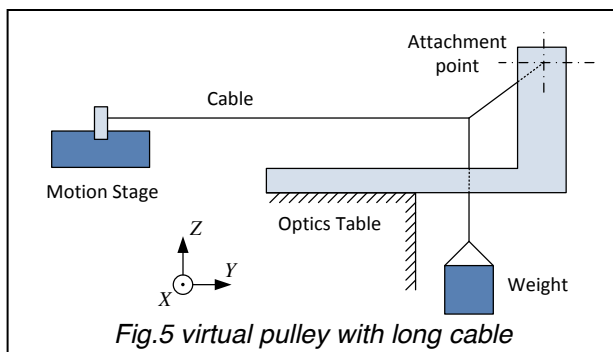
offering a satisfactory isolation from the other modes. Therefore, the dimensions in Table 1 were employed in the fabrication of the final experimental setup hardware.

Constant Force Loading Mechanism

For the SR4DoF XY mechanism shown in Fig.1, the operating point is set by applying a static Y direction force F_o . One of the challenges in applying such a constant force on the motion stage is that the latter's position varies along the X and Y axes. During the frequency response measurement, the motion stage vibrates in the X direction with a small amplitude in response to an excitation in the X direction. Also, due to cross-axis coupling, there can be an even smaller but finite Y direction vibration at the motion state in response to an X direction excitation. The Y direction actuation system has to accommodate these X and Y direction displacements during a frequency response measurement. Furthermore, since this measurement has to be repeated at various Y direction operating points ($Y_{io}=\pm 5\text{mm}$), the Y direction actuation system has to be capable of providing the necessary force (F_o) over this range and hold it constant at a given operating point.

Instead of employing an electromagnetic actuator to apply a constant force, we chose a relatively simpler actuation method utilizing steel braided cables and free weights. However, this method requires the use of a pulley to change the direction of force provided by the weights. Since a traditional pulley introduces displacement uncertainty in the loading direction due to rolling and/or sliding friction, we used a frictionless virtual pulley instead [4].

Fig.5 shows the schematic of the virtual pulley arrangement comprising three cable segments: a long horizontal segment, a vertical segment, and a diagonal segment. All three segments are connected at a common point. The other end of the horizontal segment is attached the motion stage; the other end of the vertical segment is connected to a hanging weight; and the other end of the diagonal segment is connected to a rigid extension of the ground reference (e.g. optics table). This cable arrangement is called a virtual pulley because



it converts the vertical force of the weight in the Z direction to a horizontal force F_o on the motion stage in the Y direction. The two are simply related by the trigonometry of the cable arrangement. In this manner, a constant Y direction force F_o is achieved by a known hanging weight, without the need for any active force control. The horizontal section of the cable was selected to be long enough so that it can absorb any X direction displacement of the motion stage without appreciably impacting the F_o force. Any Y direction vibration of the motion stage is also easily accommodated by the virtual pulley. The Y direction operating point is varied simply by changing the weight, and keeping track of any change in the cable arrangement trigonometry. Thus, the virtual pulley provides a frictionless pulley functionality because it eliminates any rolling or sliding interfaces and any associated uncertainties.

RESULTS AND DISCUSSION

The experimental setup hardware was fabricated and assembled for testing, as shown in Fig.6. The flexure mechanism along with the reference ground frame was made monolithically from a 25.4mm thick AL6061-T651 plate using wire-electric discharge machining (EDM), which allows dimensional tolerance of ± 0.005 mm. With the given dimensions and desired range of motion, the factor of safety against yielding was greater than 2.

The displacement of the motion stage in the X-axis was measured by a capacitance probe (Lion Precision, Range: 2 mm, Peak-peak resolution 400 nm, RMS resolution: 40 nm) mounted on a fixture attached to the ground frame. Thus the X direction displacement measurement of the motion stage with respect to the ground is not affected by the large Y direction displacement (up to ± 5 mm) of the motion stage.

A voice coil actuator (BEI Kimco Magnetics, LA24-

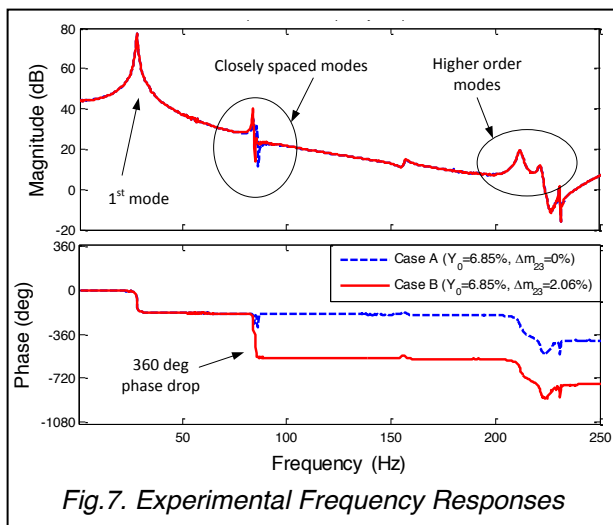


Fig. 7. Experimental Frequency Responses

20-000A, Force constant: 11.12 N/A, Stroke: ± 8.26 mm) was selected for the X direction actuation. A custom-built current amplifier based on the Power OpAmp MP111 was used to drive current in the voice coil actuator at a bandwidth of 1 kHz [1]. The system was controlled using a DSpace 1103 real-time control system with a loop-rate of 1 kHz. Frequency domain transfer function measurement was performed using a chirp signal as the input. The input and output time-domain signals were then processed using the system identification toolbox of MATLAB to obtain the frequency responses.

Fig. 7 shows the experimental frequency response from the actuator force P to the X direction displacement (X_r) of the motion stage (see Fig. 1), for two sets of operating points and mass asymmetries. The mass asymmetries are achieved by adding additional mass on the main secondary stages. Key observations are: (i) the first four experimentally measured modes match well with the above-described model; (ii) the closely spaced modes of interest are well-separated from other modes, as expected from the overall flexure mechanism design; (iii) CNMP zeros are witnessed as predicted by the prior modeling effort. For example, Case A, which corresponds to $Y_{10} = 6.85\%$ of beam length and symmetric masses (i.e. $\Delta m_{23} = 0$) shows no CNMP zeros, while Case B, which corresponds to $Y_{10} = 6.85\%$ and $\Delta m_{23} = 2.06\%$, exhibits CNMP zeros at frequencies close to the two modes associated with the secondary stages (2 and 3) of the main DPFMs.

In Fig. 2, the multiple experimental findings are overlaid on the model predictions. Note that only half of the area is tested due to the symmetric properties of the flexure with respect to positive/negative operating points. The squares indicate the conditions when CNMP zeros are present in the frequency response, while the circles indicate conditions when CNMP zeros are absent,

as measured experimentally. Fig. 2 shows that the experimental results match well with the model prediction: (i) For negative asymmetry ($\Delta m_{23} < 0$), the entire operating range is free of CNMP zeros; (ii) For $\Delta m_{23} > 0$, experimental findings largely agree with model prediction of the CNMP zero region and confirm that CNMP zeros appear only when the operating point is larger than a certain value; (iii) When nominally symmetric ($\Delta m_{23} \approx 0$), the frequency response is very sensitive to mass parameter variation in terms of the existence of CNMP zeros at large operating points.

The minor mismatch between the experimental and predicted CNMP region in Fig. 2 maybe attributed to: (i) The parameter grid resolution used in testing, and (ii) Imperfection in the flexure setup due to manufacturing and assembly tolerances.

CONCLUSION

This work has experimentally validated the previous model-based prediction [3] of the existence of CNMP zeros in the flexure mechanism of Fig. 1 for certain specific combinations of parametric asymmetry and operating point. Significantly, this work shows that via an intentionally asymmetric choice of secondary stage masses, one can completely eliminate the CNMP zeros, thereby enabling better closed-loop dynamic performance. Furthermore, since the flexure beams are still symmetric, the quasi-static performance (which is independent of masses) in terms of cross-axis decoupling and error motions at the motion stage can still be preserved.

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NON-COLLOCATED OPTIMAL CONTROL OF FLEXIBLE DRIVES WITH LOAD SIDE FEEDBACK

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INTRODUCTION

This paper presents a novel optimal motion control technique for feed drive systems with structural flexibility. The structural resonance is modeled and actively compensated within the controller structure utilizing only the load side feedback, i.e. in a non-collocated fashion. Controller gains are tuned to inject modal damping and attain high positioning bandwidth with vibration suppression. Mainstream industrial controllers rely on the convenient P-PI cascade control structure to achieve moderate positioning performance [1]. However, structural flexibilities limit their performance. For instance, high gain P-PI controllers implemented with motor side feedback (collocated operation) may cause severe vibrations on the load side. On the other hand, if they are implemented with load side feedback (non-collocated operation) they suffer from instability. As a result, cascade P-PI controller structure is not a viable solution for accurate, high-bandwidth control of high speed feed drive systems [2], [3]. In order to cope with that, this work utilizes load side position, velocity, acceleration and jerk measurements to control the structural dynamics of the flexible drive. The pole locations are assigned optimally based on the Linear Quadratic Regulator (LQR) framework by weighting modal damping and tracking objectives [5].

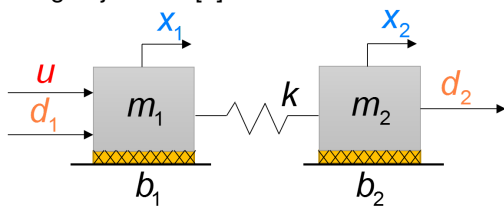


FIGURE 1. Free body diagram of a typical flexible feed drive system.

SYSTEM MODELING AND CONTROL LAW

Typically, a flexible feed drive system can be modeled as a 2DOF system based on the free body diagram shown in Fig .1. The governing set of differential equations can be derived as

$$\begin{aligned} m_1 \ddot{x}_1 &= -b_1 \dot{x}_1 + k(x_2 - x_1) + u \\ m_2 \ddot{x}_2 &= -b_2 \dot{x}_2 + k(x_1 - x_2) \end{aligned} \quad (1)$$

where x_1 and x_2 are displacements of the masses m_1 and m_2 . m_1 denotes the mass of the motor and m_2 denotes mass of the load/table. u is the motor force, or so-called the equivalent control input that drives the system. b_1 and b_2 are viscous friction forces and k is the equivalent spring constant representing the flexibility in-between masses, i.e. motor and table. Typically, flexibilities in a feed drive can be originated from any structural element in the feed drive system. For instance, the screw-nut interface introduces a lightly damped flexibility in most ball-screw driven motion systems [3].

Equation (1) can be re-written to express the dynamics of the table position x_2 with respect to the control input, u directly as:

$$\frac{k}{m_1 m_2} u = x_2^{(iv)} + a_3 \ddot{x}_2 + a_2 \dot{x}_2 + a_1 \dot{x}_2 \quad (2)$$

where

$$\begin{aligned} a_1 &= \frac{(b_1 + b_2)k}{m_1 m_2}, \quad a_2 = \frac{b_1 b_2 + (m_1 + m_2)k}{m_1 m_2}, \\ a_3 &= \frac{b_1 m_2 + b_2 m_1}{m_1 m_2} \end{aligned} \quad (3).$$

The 4th order differential equation presented in (2) can be re-written as a set of 1st order differential equations, i.e. in state space form, by selecting the state vector as higher order derivatives of the table position

$\mathbf{v} = [x_2 \quad \dot{x}_2 \quad \ddot{x}_2 \quad \ddot{x}_2]^\top$ as follows

$$\begin{bmatrix} \dot{x}_2 \\ \ddot{x}_2 \\ \ddot{x}_2 \\ x_2^{(iv)} \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & -a_1 & -a_2 & -a_3 \end{bmatrix}}_{\Lambda} \underbrace{\begin{bmatrix} x_2 \\ \dot{x}_2 \\ \ddot{x}_2 \\ \ddot{x}_2 \end{bmatrix}}_{\mathbf{v}} + \underbrace{\begin{bmatrix} 0 \\ 0 \\ 0 \\ k/m_1 m_2 \end{bmatrix}}_{\beta} u \quad (4)$$

Note that the representation in (4) is in controllable canonical form and the eigenvalues of the system can be computed as the roots of the 4th order polynomial,

$$s(s^3 + a_3 s^2 + a_2 s + a_1) = 0 \quad \text{or} \quad (5)$$

$$s(s + \lambda)(s^2 + 2\zeta\omega_n s + \omega_n^2) = 0$$

which are the eigenvalues of $\mathbf{\Lambda}$,

$$\text{eig}(\mathbf{\Lambda}) = \begin{cases} p_1 = 0 \\ p_2 = -\lambda \\ p_{3,4} = -\zeta\omega_n \pm j\omega_n\sqrt{1-\zeta^2} \end{cases} \quad (6).$$

The 4 eigenvalues in (6) govern the rigid body and the vibratory dynamics. p_1 is the integral action that converts table velocity to table position, p_2 (or $-\lambda$) is the rigid body pole and p_3 and p_4 are the complex conjugate vibratory poles with the damping ratio ζ and natural frequency ω_n .

Furthermore, based on the state space representation given in (4), a pole placement controller can be designed to locate all the poles at desired locations and realize direct feedback control by simply measuring the load side position x_2 , velocity $\dot{x}_2$, acceleration $\ddot{x}_2$ and jerk $\ddot{\ddot{x}}_2$. In other words, a state feedback control law can be written as

$$u = \mathbf{K}_{1 \times 4}(\mathbf{v}_r - \mathbf{v})$$

$$= k_p(x_{2r} - x_2) + k_v(\dot{x}_{2r} - \dot{x}_2) \quad (7)$$

$$+ k_a(\ddot{x}_{2r} - \ddot{x}_2) + k_j(\ddot{\ddot{x}}_{2r} - \ddot{\ddot{x}}_2)$$

where $\mathbf{v}_r = [x_{2r} \ \dot{x}_{2r} \ \ddot{x}_{2r} \ \ddot{\ddot{x}}_{2r}]^T$ is the reference state vector containing higher order derivatives of the trajectory, and $\mathbf{K} = [k_p, k_v, k_a, k_j]$ contains feedback gains, which can be computed based on the desired dynamics using conventional pole assignment techniques such as Ackermann's method [4].

Thus, by measuring the table position, velocity, acceleration and jerk dynamics of feed drive can be designed freely. In other words, the positioning bandwidth as well as damping of the vibration can be controlled [6]. However, such a controller is inherently not robust against disturbances and un-modelled dynamics. This can be tackled by introducing integral action to the feedback controller. Original state space representation from (4) is augmented with an integral state,

$$\begin{bmatrix} \dot{\mathbf{v}} \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} \mathbf{\Lambda} & \mathbf{0}_{4 \times 1} \\ 1 & \mathbf{0}_{1 \times 4} \end{bmatrix} \begin{bmatrix} \mathbf{v} \\ \int_0^t x_2(\tau) d\tau \end{bmatrix} + \begin{bmatrix} \boldsymbol{\beta} \\ 0 \end{bmatrix} u \quad (8)$$

$$\begin{matrix} \mathbf{x} & \mathbf{A} & \mathbf{x} & \mathbf{B} \end{matrix}$$

and the control law is updated as,

$$u = \mathbf{K}_{1 \times 5}(\mathbf{x}_{r2} - \mathbf{x})$$

$$= k_p(x_{2r} - x_2) + k_v(\dot{x}_{2r} - \dot{x}_2)$$

$$+ k_a(\ddot{x}_{2r} - \ddot{x}_2) + k_j(\ddot{\ddot{x}}_{2r} - \ddot{\ddot{x}}_2) \quad (9).$$

$$+ k_i \int_0^t (x_{2r}(\tau) - x_2(\tau)) d\tau$$

As observed, above control law can be implemented as a PID action on the position feedback with a parallel PD loop operating acceleration feedback in industrial controllers.

OPTIMAL POLE ASSIGNMENT

The basic pole placement controller presented in the previous section can be designed by carefully assigning all the 5 closed loop poles associated with the rigid body and vibratory dynamics. Typically, closed loop poles of flexible feed drive systems are determined either based on heuristics, approximations [7] or through engineer's expertise. The goal in tuning most industrial feed drive systems is to achieve good tracking and disturbance rejection and the same time dampen structural vibrations for precision positioning. Although these objectives are not directly contradicting with each other, significant control action is necessary to achieve both of them, which in return demands large actuation forces and may cause saturation. The objective in this section is to determine optimal closed loop pole locations to balance the feed drive performance for tracking, vibration suppression and control effort.

An optimal control strategy that seeks a balance between these objectives can be designed using the Linear Quadratic Regulator (LQR) framework [5]. The problem is to find the optimal pole locations which minimizes the following performance index

$$J = \int_0^\infty (\mathbf{x}^T \mathbf{Q} \mathbf{x} + \mathbf{u}^T \mathbf{R} \mathbf{u}) dt \quad (10).$$

The first term in (10) penalizes deviation of the states from a set point and second term penalizes the control action. Here, $\mathbf{Q}$ is the constant weight matrix, which increases contribution of each state, and $\mathbf{R}$ weights the control effort. Notice that since the pole placement problem given in (4) is single input multiple output, $\mathbf{R}$ becomes a scalar gain. However, design of the $\mathbf{Q}$ matrix is not trivial to achieve tracking and vibration suppression at the same time. The following analysis is undertaken to design the state weight matrix $\mathbf{Q}$.

The first objective is to decide on which state must be penalized in the performance index to suppress structural vibrations.

Firstly, assuming that viscous friction is small in precision feed drive systems, $b_1=b_2=0$, (2) is simplified as follows

$$\underbrace{x_2^{(iv)} + \frac{k(m_1 + m_2)}{m_1 m_2} \ddot{x}_2}_{\omega_n^2} = \frac{k}{m_1 m_2} u \quad (11).$$

which clearly shows dynamics of the undamped vibratory mode. Next, the following control law based on sole jerk feedback ($\ddot{x}_2$) is postulated to inject damping into the system

$$u = -\phi \ddot{x}_2, \quad \phi \in \mathbb{R} \quad (12).$$

Substituting (12) in (11) yields

$$x_2^{(iv)} + \underbrace{\phi \frac{k}{m_1 m_2} \ddot{x}_2}_{2\zeta\omega_n} + \underbrace{\frac{k(m_1 + m_2)}{m_1 m_2} \ddot{x}_2}_{\omega_n^2} = 0 \quad (13).$$

As shown in (13), feeding back table jerk ($\ddot{x}_2$) directly influences vibratory dynamics by injecting damping. In other words, minimizing deviation of the jerk of the load from the set-point acts as dampening effect on the structural vibrations, and linearly increases damping of the resonant mode, e.g. $\zeta^{new} = \phi \zeta^{old}$. This indicates that the performance index in (10) should penalize the jerk state of the table for robust vibration damping.

As observed from (13), minimizing jerk of the table does not affect velocity dynamics (i.e. rigid body dynamics) of the drive system. In other words, static compliance or the tracking bandwidth of the feed drive system are not altered. Nevertheless, tracking and disturbance rejection are the other performance pillars in motion control. Particularly, integral of the load

side position ($\int_0^t x_2(\tau) d\tau$) is an indicator for

tracking and low frequency disturbance rejection performance. Thus, the performance index in (14) should also generate integral action, and hence by defining $\mathbf{H} = [0 \ 0 \ 0 \ \alpha \ 1]$ both jerk and integral of the table are penalized in (10) as:

$$\begin{aligned} J &= \int_0^\infty (\rho \mathbf{x}^T \underbrace{(\mathbf{H}^T \mathbf{H})}_{\mathbf{Q}} \mathbf{x} + u) dt \\ &= \int_0^\infty \left(\rho \left(\underbrace{\alpha \ddot{x}_2 + \int_0^t x_2(\tau) d\tau}_{\mathbf{z}} \right)^2 + u \right) dt \end{aligned} \quad (14).$$

Notice that $\mathbf{z}$ is a state composed of weighted linear combination of table jerk and integral states. This formulation allows us to represent the system as a SISO form where u is the input and $\mathbf{z}$ is the output. α is a scalar weight and becomes a tuning parameter to balance between vibration suppression and tracking. ρ is the second tuning parameter of the proposed controller, which weights control action, i.e. control effort, over control performance.

Finally, the optimal state feedback gains $\mathbf{K}_{1 \times 5}$ that minimizes the performance index J is obtained by solving the Algebraic Riccati Equation defined by $\mathbf{A}$, $\mathbf{B}$, ρ and α in the LQR framework [5], and implemented in the form of (9) as a $\text{PID}_{pos} + \text{PD}_{acc}$ type feedback controller.

EXPERIMENTAL SETUP AND CONTROLLER PERFORMANCE

Previous section presented the proposed optimal motion controller scheme. This section validates performance and practicality of the controller experimentally. The experimental 2DOF linear motor driven drive system is shown in Fig. 2. The cart on the left hand side is driven by a linear motor and connected to a load on the right hand side with a pair of linear springs. Both carts are equipped with resolvers with 4 [μm] resolution. The servo amplifier is set to operate on torque control and the control signal is generated at a real-time computer at 1 kHz.

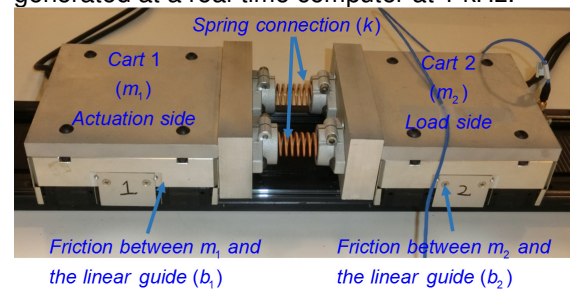


FIGURE 2. Experimental Setup.

Firstly, system dynamics is identified through transfer function fitting in frequency domain. Fig. 3 shows actual measured frequency responses from both carts, namely x_1 and x_2 . Identified transfer functions between control signal to cart positions are:

$$\frac{X_1(s)}{U(s)} = \frac{2671(s^2 + 7.881s + 6791)}{s(s + 2.752)(s^2 + 9.872s + 13700)} \quad (15)$$

$$\frac{X_2(s)}{U(s)} = \frac{1.875(10^7)}{s(s + 2.752)(s^2 + 9.872s + 13700)} \quad (16)$$

Due to rough resolution of resolvers, an accelerometer with a sensitivity of 102.0 mV/m/s² is attached to the second cart, and it is used to measure acceleration and jerk states. In the actual implementation, a reduced order observer is designed to realize high bandwidth acceleration and jerk state measurements.

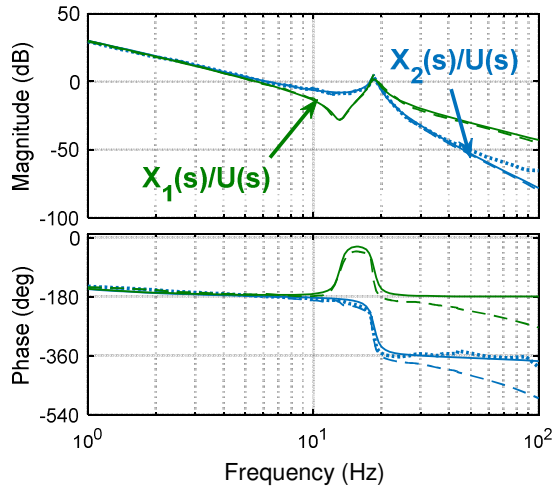


FIGURE 3. FRFs for ideal models (solid lines), encoder measurement (dashed lines) and accelerometer measurement (dotted line, only for 2nd cart) regarding the experimental setup.

As observed from Fig. 3, the system has a vibratory mode at 117 rad/s (18.62 Hz) with 4.2% damping and a rigid body mode at 2.752 rad/s (0.438 Hz). Firstly, parameters of the proposed controller are tuned for moderate control performance as $\alpha=0.292(10^{-3})$ and $\rho=3(10^3)$. The rationale behind the selection of α and ρ will be discussed in the following section. Proposed controller is benchmarked against the conventional P-PI controller with motor side velocity and table side position feedback loops. The P-PI control law 3 tunable gains yields;

$$u_{PPI} = K_V [(x_{2r} - x_2)K_P + \dot{x}_{2r} - \dot{x}_1] + K_I \int_0^t [(x_{2r}(\tau) - x_2(\tau))K_P + \dot{x}_{2r}(\tau) - \dot{x}_1(\tau)] d\tau \quad (17)$$

K_V and K_I are proportional and integral velocity gains and K_P is the table-side position feedback gain. These 3 parameters only allow placement of 3 closed loop poles. As opposed, proposed controller can place 5 of the closed loop poles freely. In order to compare the P-PI controller performance to the proposed controller fairly, 2 of these poles are chosen to match velocity dynamics poles of the proposed controller. The 3rd pole is chosen to be same as the rigid body

pole of the open loop system and compensated by the PI velocity loop. The remaining 2 poles became residuals that are self-assigned. Such an arrangement yields comparable gain cross-over frequencies and phase margins for the experimental setup. Table 1 summarizes the pole locations, controller parameters and frequency domain characteristics of P-PI and proposed optimal controller.

TABLE 1. Closed loop poles, controller parameters and frequency domain characteristics of proposed and P-PI controllers.

	Proposed		PPI	
Closed loop poles [rad/s]	$-23.3 \pm 32.7i$		$-23.3 \pm 32.7i$	
	$-45.7 \pm 110i$		$-23.3 \pm 101i$ (residual)	
	-44.4		-2.752	
Controller Parameters	k_p	3.1635	K_V	0.0298
	k_v	0.089		
	k_a	$6.72(10^{-4})$	K_I	0.082
	k_j	$9.55(10^{-6})$		
	k_i	54.7723	K_P	29.6242
Gain cross-over freq. [Hz]	28.50		23.87	
Phase Margin [deg.]	41.6		45.7	

Fig. 4 represents loop transmission and the Nyquist plots for the controllers. Notice that while proposed controller has slightly higher cross-over frequency, P-PI controller shows a slightly larger phase margin. Fig. 5 shows experimental tracking performance comparison on a jerk limited point-to-point trajectory. As shown, proposed controller tracks the reference trajectory more accurately and with less vibrations. Fig. 6 gives more insight on the controller. As observed from the tracking and load side disturbance frequency responses, proposed controller shows much larger tracking bandwidth and much greater disturbance rejection capabilities. Moreover, the residual closed loop poles of the P-PI controller are slower than the damped oscillatory poles of the proposed controller. This lowers the disturbance performance of P-PI controller.

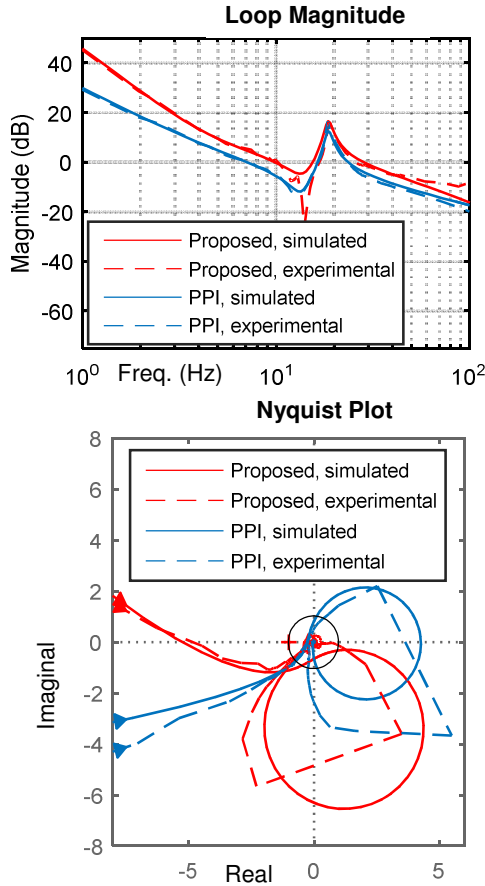


FIGURE 4. Loop transmission characteristics for both controllers in frequency domain.

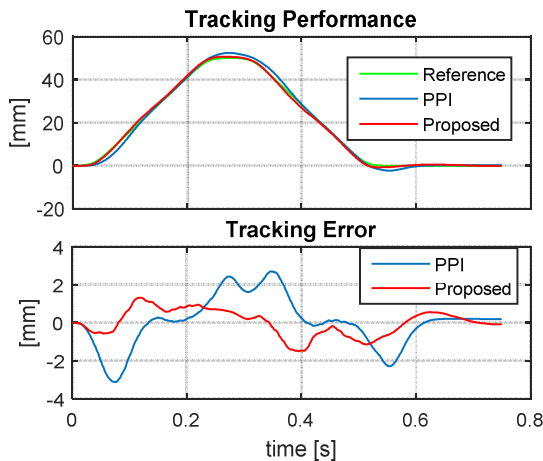


FIGURE 5. Tracking performance of proposed and P-PI controllers.

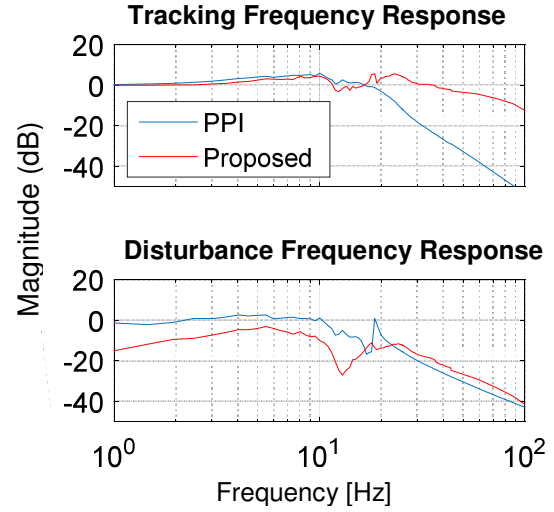


FIGURE 6. Tracking and load disturbance rejection for proposed and PPI controllers.

CONTROLLER ANALYSIS

Proposed controller has only two tuning parameters, namely α and ρ . This section analyses effect of controller parameters and presents tuning guidelines. In the proposed controller, α gain weights damping over tracking, and ρ weights the control action over control performance. The trajectory of the closed loop pole locations can be obtained using the symmetric root locus technique [5] by adding right hand side complementary poles ($-s$) to the closed loop system as

$$\Pi(s) = 1 + \rho \frac{[\mathbf{H}(-s\mathbf{I} - \mathbf{A})^{-1}\mathbf{B}]^T [\mathbf{H}(s\mathbf{I} - \mathbf{A})^{-1}\mathbf{B}]}{G^*(s)} \quad (18)$$

where $G^*(s)$ is the open loop transfer function with mirrored poles and additional zeros controlled by α . Considering only left hand side poles, (18) presents optimal closed loop pole locations with varying ρ . $G^*(s)$ can be represented based on (16)

$$G^*(s) = \rho \underbrace{(s^8 \alpha^2 + 2s^4 \alpha + 1)}_{N(s)} \frac{X_2(-s)}{U(-s)} \frac{X_2(s)}{U(s)} \quad (19)$$

which shows that G^* has 8 zeros (4 repeated zero couples) that are controlled by α gain as

$$s_m = \frac{1}{\sqrt[4]{\alpha}} e^{-j \frac{(2m-1)\pi}{4}}, \quad -1 \leq m \leq 2, \quad m \in \mathbb{Z} \quad (20).$$

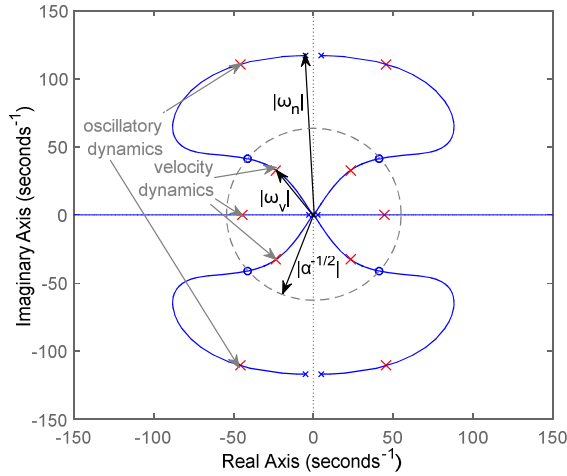


FIGURE 7. Symmetric root locus and optimal closed loop poles (shown in red, on the left half plane).

Fig. 7 illustrates symmetric root locus of the system based on (18). Optimal closed loop poles are the ones that are on the left half side of the complex plane. The trajectories of the closed loop poles are controlled by α . In other words, since increasing ρ simply moves closed loop poles to the zeros of G^* , bandwidth of the system is guided by α . α should be selected moderately larger than the desired natural frequency of underdamped closed loop rigid body poles, ω_v . Notice that, the final tracking bandwidth of the system will be larger than ω_v due to feed-forward terms in (9). On the other hand, assuming that the resonant mode frequency ω_n is higher than ω_v , α should be selected as

$$\omega_n > \frac{1}{\sqrt[2]{\alpha}} > \omega_v \quad (21)$$

Based on the analysis, following guideline is proposed for effective tuning of the controllers. Firstly, natural frequency of underdamped closed loop rigid body poles, ω_v , is selected as one third of the resonant frequency of the feed drive system as $\omega_v = \frac{\omega_n}{3}$. Then, α is tuned to place zeros of G^* at the half of the resonant mode $\frac{1}{\sqrt[2]{\alpha}} = \frac{\omega_n}{2}$. Finally, ρ is increased until the desired value of ω_v is reached on the symmetric root locus.

CONCLUSION

This paper has presented a high order non-collocated closed loop controller for flexible feed drives. Proposed controller is formulized as a state feedback pole placement controller with an additional integral state using only load side information. The decision for closed loop pole locations are made by defining an optimal control strategy that takes control effort, tracking, disturbance rejection and vibration suppression performances into account. The resulting pole placement strategy reduces the decision making process for 5 closed loop poles into tuning 2 optimal control parameters. Experimental results show that the designed controller outperforms industrial P-PI controllers with a single feedback sensor and the tuning is significantly intuitive.

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PRECISE MEASUREMENT AND CONTROL OF THE BLOWN-AIR FLOW RATE FOR A ROBOT PLAYING A WIND INSTRUMENT SUCH AS A RECORDER OR KHLUI

-FUNDAMENTAL EXPERIMENT FOR EXPRESSING TREMOLO-

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INTRODUCTION

In the last few decades, many types of robots have been developed for various purposes. While most robots are used in industrial and manufacturing applications, some robots are used in fields other than manufacturing, such as rescue activities [1] and medical surgeries [2]. Robots for entertaining people have also been reported [3].

As part of the research for the development of robots that entertain people, the authors have been developing robots that can play musical wind instruments, such as soprano and alto recorders and khlui (traditional Thai vertical duct flute) [4][5][6]. The aim of the current authors' research has been to construct a robot that can play such wind instruments with natural and realistic sounds, as if an expert human musician were playing the instrument. To meet this aim, the authors have been applying pneumatic unsteady flow rate measurements and control techniques in a robot system.

In this paper, first, the overview of the developed robot system is presented. Second, the flow sensor and the method of calibrating the dynamic characteristics of the sensor are introduced. Third, a fundamental experiment using the dynamically calibrated flow sensor for a robot expressing tremolo is shown.

OVERVIEW OF THE DEVELOPED ROBOT SYSTEM

We have been developing the recorder-playing robot shown in Figure 1 and Figure 2. The robot system consists of a computer, a musical keyboard (Edirol musical instrument digital interface (MIDI) keyboard controller PC-50), an

electronic circuit as the signal receiver and fingering controller, an SP valve (Festo MPYE-M5-B-SA), a soprano recorder (Aulos 503B), and a fingering part consisting of solenoid plungers (Yamaha DC Solenoid MD-232). In the computer, MIDI 3 sequencer software (Cakewalk SONAR 6 LE) generates MIDI signals for playing (controlling) the recorder. The sequencer software also generates accompaniment music in synchronization with the MIDI signals. The MIDI signal is sent to the MIDI signal transmitter. Then, the MIDI signal is received by the electronic circuit and divided into two signals. One signal controls the blown air and the other controls the fingering part. Eleven solenoid plungers are attached to acrylic hinge plates as robot fingers. In addition to the recorder playing robot, the authors have fabricated the khlui playing robot shown in Figure 3. The khlui robot was developed as an international collaboration. The robot even played on a stage at the Engineering Expo 2015 in Bangkok [6].

FLOW SENSOR AND DYNAMIC CALIBRATION

Quick Response Laminar Flow Sensor

To measure and control the blown air for a recorder or khlui, a flow sensor having sufficient resolution and dynamic characteristics is needed. The type of flow sensor used in this research is a laminar flow type, named "quick response laminar flow sensor" (QFS), which our research group has been developing [7]. As shown in Figure 4, the QFS is composed of a laminar flow element and a differential pressure gauge.

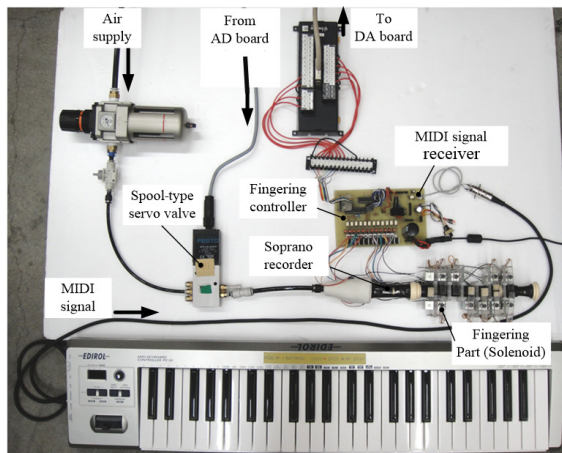


Figure 1. Recorder-playing robot.

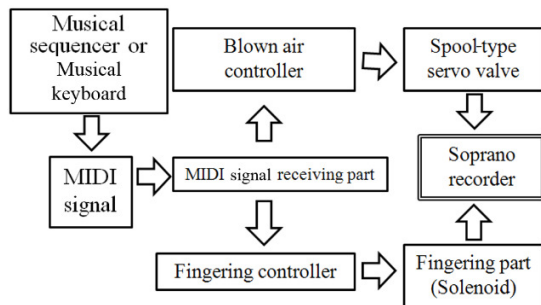


Figure 2. Configuration of recorder playing robot.

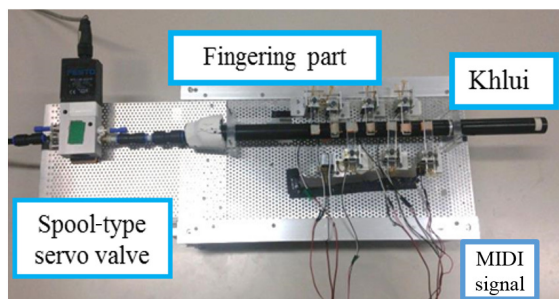


Figure 3. Khlui playing robot.

The model of the QFS used in this research is QFS-0.3-50-30 (Tokyo Meter Co., Ltd.). As an example, if the resolution of the differential pressure gauge is 0.1 Pa, the Ambiguous Name Resolution (ANR) of the QFS is 25.6 mL/min.

Dynamic Calibration of QFS Using an Unsteady Flow Generator

The dynamic characteristics of the QFS up to 20 Hz were tested by using an unsteady flow generator (UFG).

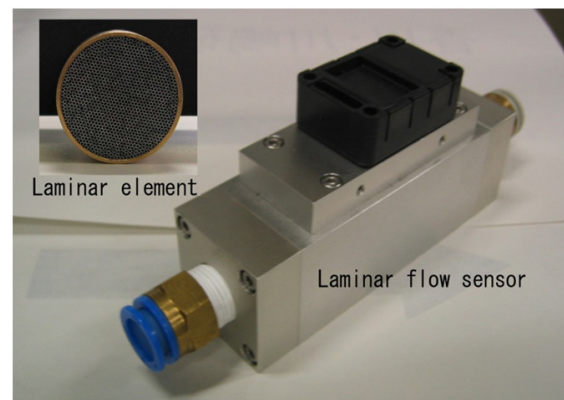


Figure 4. Quick response laminar flow sensor (QFS).

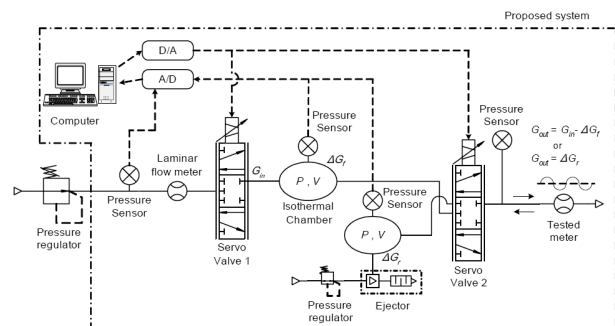


Figure 5. Schematic of unsteady flow generator (UFG).

The UFG is a device that can generate arbitrary oscillation of the air flow up to 50 Hz at least [8]. A schematic of the UFG is shown in Figure 5. The UFG includes two spool-type valves and an isothermal chamber. A schematic of the test of the dynamic characteristics of the QFS is shown in Figure 6. The QFS is set downstream from the UFG, which is open to the atmosphere. Both the generated flow rate from the UFG (as the standard) and the measured flow rate using the QFS are recorded and compared.

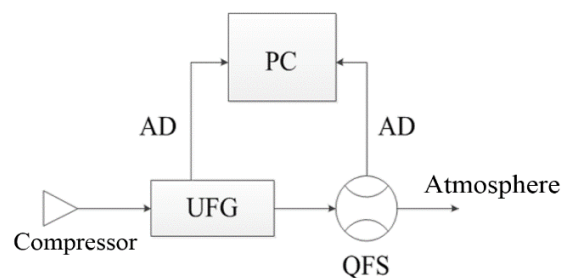


Figure 6. Schematic of dynamic characteristics test.

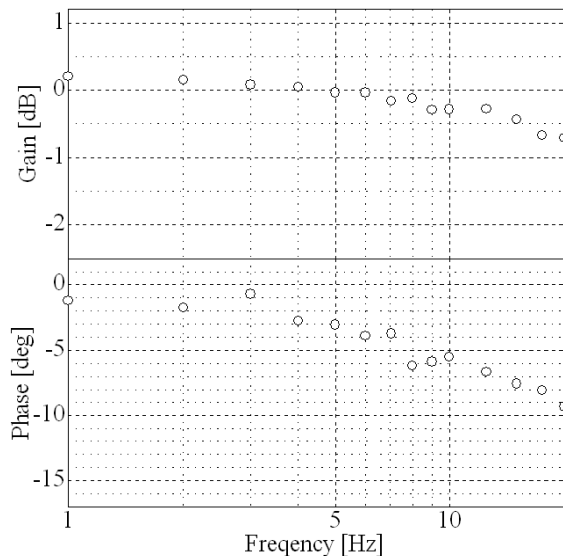


Figure 7. Bode diagram of dynamic characteristics of the QFS.

In the experiments, the set value of the generated flow rate from the UFG is defined according to equation (1).

$$G \text{ [g/s]} = 0.216 + 0.108 \sin(2\pi ft) \quad (1)$$

The experimental results are summarized in the Bode diagram shown in Figure 7. In the Bode diagram, the generated flow rate using the UFG is the denominator and the measured flow rate using the QFS is the numerator. The experimental results show that when $f = 20$ Hz, the gain is -0.8 dB, and the phase is -9 deg. The needed frequency range for measuring the blown air into a recorder or a khlui is several hertz, so the QFS is considered to be suitable for the measurement of blown air.

FUNDAMENTAL EXPERIMENT FOR EXPRESSING TREMOLO WITH ROBOT USING QFS

Why Expressing Tremolo is Difficult

Although the authors have been trying to realize special musical sound effects, such as tonguing, vibrato, and tremolo, with the robot system, the task remains difficult. In this paper, a method for expressing proper tremolo is discussed.

Tremolo is usually defined as a trembling sound effect. Here, the authors define tremolo as a variation of the amplitude, that is, the volume of the sound changes sinusoidally while keeping the musical interval constant.

One reason why it is difficult for a recorder or khlui playing robot to express tremolo is that making the blown-air flow rate to sinusoidally change the sound volume requires changing the musical interval.

However, it is not impossible to express tremolo with a recorder or khlui. The authors carefully watched a movie of an expert musician playing the khlui and found that the player expresses tremolo not only with blown-air control but also with subtle fingering using both soft and tight pressure on the tone holes. Thus, it seems that combining blown-air flow rate control with force control of the fingering part can express tremolo properly and naturally.

Experimental Procedure and Results

To validate our hypothesis stated at the end of the previous section regarding how to express tremolo, a fundamental experiment using the QFS was conducted. The experimental setup is shown in Figure 8 and Figure 9. In the setup, air is supplied from a compressor to a regulator, and then the flow rate is adjusted by a speed controller (SMC AS2000). The air flow rate supplied to the recorder is measured by the QFS. The flow rate Q [L/min (ANR)] is first adjusted to 6.9 L/min (ANR), which is the optimum flow rate for expressing A5 (880 Hz).

At the same time, the fingering force is adjusted by using a lead screw, and the force F_p [N] is measured by a load cell (Tokyo-sokki TCLZ-50NA).

In the experiment, the tone pitch is measured by a tuner (KORG GA-1) and the sound pressure level P_s [dB] is measured by a sound level meter (Custom Corporation MRS-1).

In the experiment, the flow rate Q was changed from 6.9 L/min (ANR) to 4.4 L/min (ANR) at intervals of 0.5 L/min (ANR). When Q is decreased, the sound tone pitch (frequency f [Hz]) is decreased. However, even if Q is decreased, if F_p is also decreased, then f can be kept constant, which we did for each Q .

The experimental results of F_p , P_s , and f are shown in Figure 10 (plots show averages and standard deviations over 5 repetitions of the experiment). As shown, with decreasing Q and decreasing F_p , sound having the same f with various P values (i.e., sound volume) was realized, which indicates that combining control of the blown-air flow rate and control of the fingering force in the robot system is a promising approach for a robot to express natural and proper tremolo, as if an experience human musician were playing a recorder or khlui.

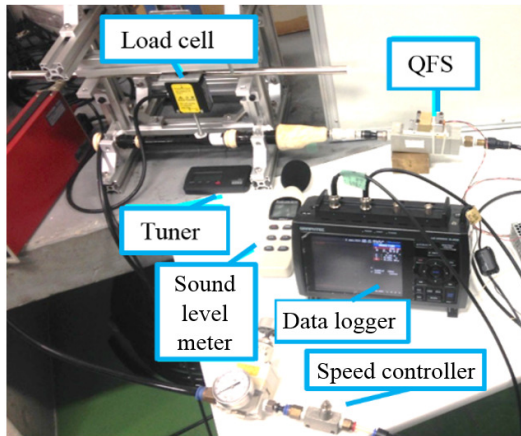


Figure 8. Experimental setup.

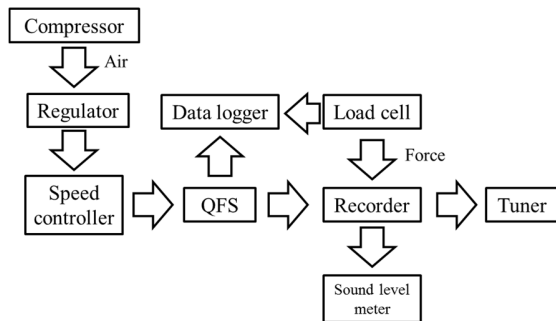


Figure 9. Schematic of experimental setup.

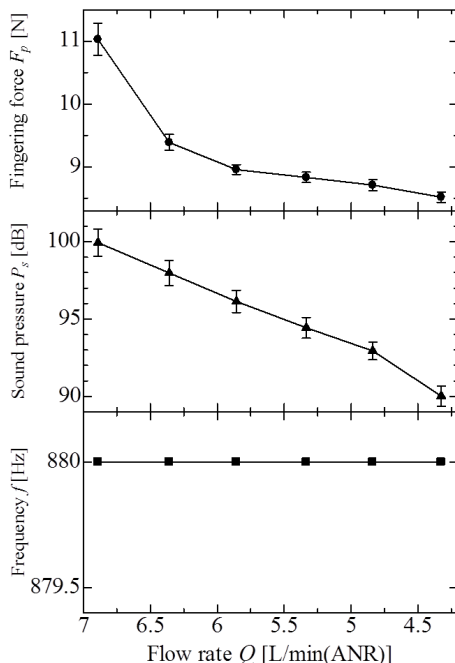


Figure 10. Experimental results.

CONCLUSIONS

In this paper, an overview of a developed robot system is explained. Then, a flow sensor and the method of calibrating its dynamic characteristics are introduced. Finally, a fundamental experiment using a dynamically calibrated flow sensor for a robot expressing tremolo is presented.

As future work, the method will be applied to a robot system to express natural and proper tremolo.

ACKNOWLEDGEMENT

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Development of a Spherical Motor Manipulated by Four Wires

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INSTRUCTIONS

Recently, industrial robots can perform various tasks which are precise assemblies with ultra-precise positioning, delicate assemblies with force control, or intelligent assemblies using visual information. However, although the programmable performance of the industrial robots has progressed higher, structure of the industrial robots is still multi-link mechanism which is the same as early industrial robots. Therefore, if the industrial robot has a spherical joint like a shoulder joint or a wrist joint of human, we can simply develop control programs without considering interference of the multi-links mechanisms; also, formations of the industrial robots become smart and slim.

Several spherical motors have been developed for getting these advantages ([1] Kaneko et al. 1989; [2] Toyama et al. 1996). One spherical motor composes a special planet gear mechanism. The other spherical motor consists of a spherical body and four ultrasonic motors. However, these spherical motors are still complicated, and it is difficult to make those control programs. For getting simple structure and smart programs, we are developed a novel spherical motor manipulated by four wires [3]. In this paper, movable range of the spherical motor is theoretically analyzed and confirmed by experiments.

STRUCTURE OF THE SPHERICAL MOTOR

The spherical motor consists of a spherical body, a pedestal with a spherical concavity, and four wires as shown in Figure 1 and Figure 2. The spherical body is fitted on the spherical concavity of the pedestal, and smoothly slides to three directions in rotation of θ_x , θ_y , and θ_z . Four wires are hung at two pins on the spherical body, and these wires are guided into two small holes on the rim of the spherical concavity. Each wire is tightened between its hanging pin and its guiding hole on the surface of the spherical body. Four wires are pulled or are loosened by stepping motors as shown in Figure 2.

The lengths of the tightened wires are uniquely decided against any posture of the spherical body. When the wire lengths are

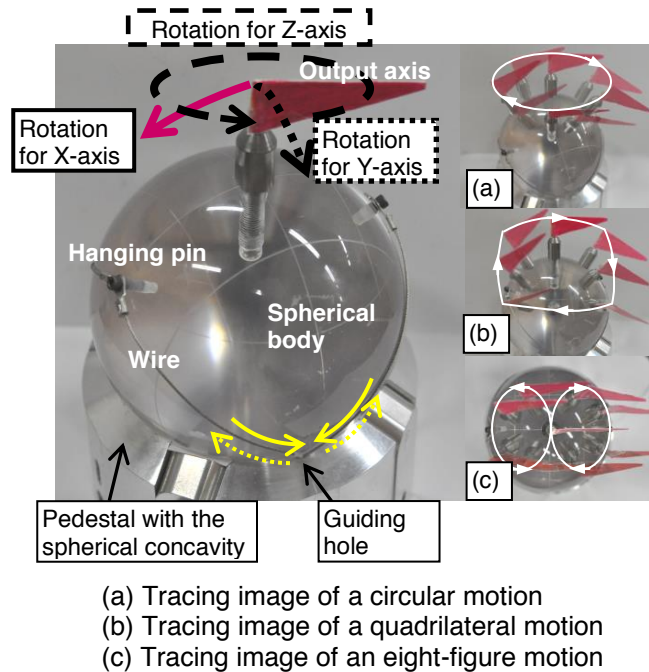


FIGURE 1. Motion of the spherical motor manipulated by four wires

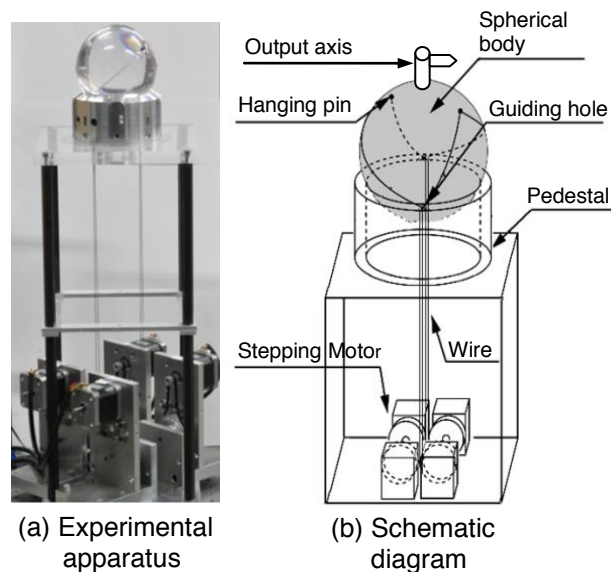


FIGURE 2. Structure of the spherical motor manipulated by four wires

controlled to the calculated length, the spherical body is driven to three directions in rotation of θ_x , θ_y , and θ_z as shown in Figure 1. Additionally, if the lengths of the wires are continuously controlled to calculated length, the spherical body is moved into the circular motion, the quadrilateral motion, or the eight-figure motion.

DRIVE PRINCIPLE OF THE SPHERICAL MOTOR

The spherical motor is driven as shown in Figure 3. Coordinate system and standard posture of the spherical motor are defined as shown in Figure 3(a). If the wires A and B are pulled or are loosened; and also, if the wires C and D are loosened or are pulled, the spherical motor rotates about the X-axis as shown in Figure 3(b). If the wires A and D are pulled or are loosened; and also, if the wires B and C are loosened or are pulled, the spherical motor rotates about the Y-axis as shown Figure 3(c). If the wires A and C are pulled or are loosened; and also, if the wires B and D are loosened or are pulled, the spherical motor rotates about the Z-axis as shown in Figure 3(d).

THEORETICAL ANALYSIS OF POSSIBLE / IMPOSSIBLE RANGE FOR MOVING

Figure 4(a) shows the analysis model of the spherical motor in the standard posture. Point P1 and P2 indicate the hanging pins, and Q1 and Q2 indicate the guiding holes. Coordinate frame of (x, y, z) is fixed against the pedestal, and the origin of the coordinate frame get to the center of the spherical concavity of the pedestal. In the standard posture, Point P1 takes at latitude 30° north and longitude 0° east on the spherical body. Similarly, Point P2 takes at latitude 30° north and longitude 180° east. Point Q1 takes at latitude 30° south and longitude 90° east, and Point Q2 take at latitude 30° south and longitude 90° west.

In theoretical analysis, firstly, the spherical body is turned with ω as shown in Figure 4(b). Secondly, the spherical body is inclined with β to direction of longitude of θ . Posture of the spherical body is defined by (ω , θ , β).

When the spherical body is turned with ω and is inclined to the direction of the longitude of θ , the maximum incline angle of $\beta_{\max}$ are analyzed. If the incline angle exceeds the maximum incline angle of $\beta_{\max}$, the spherical body cannot continue to incline, or the spherical body cannot go back to the standard posture, or the spherical body cannot move to the direction

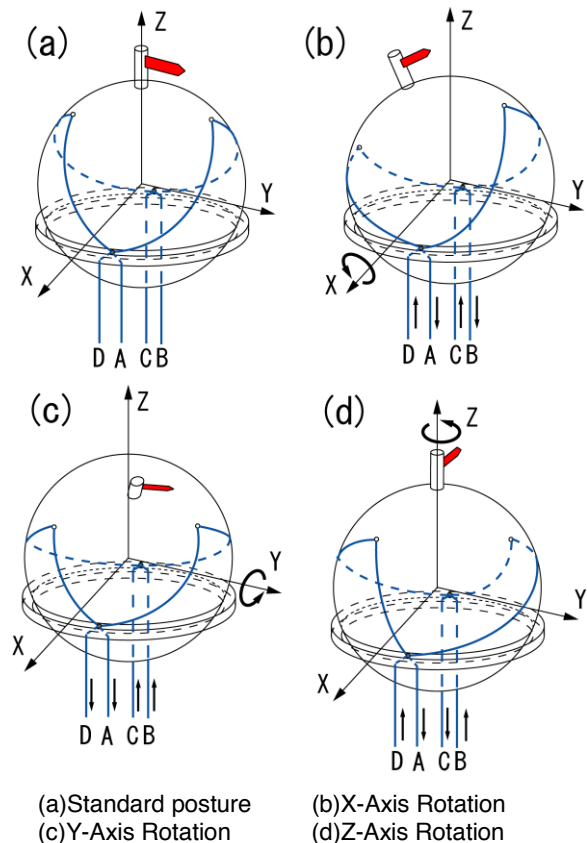


FIGURE 3. Drive principle of the spherical motor manipulated by four wires

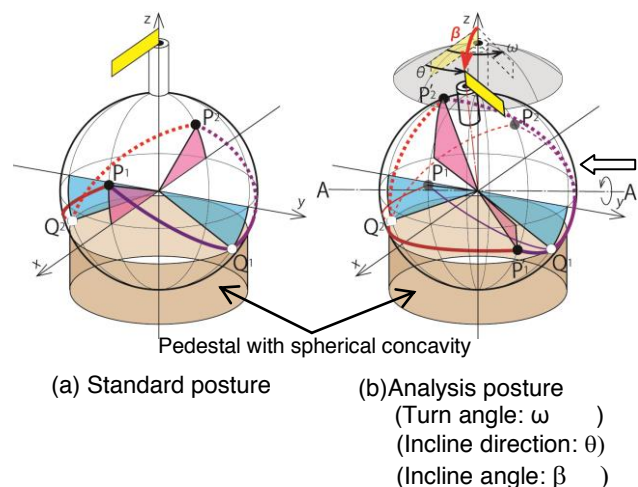


FIGURE 4. Theoretical analysis model of the spherical motor

perpendicular to the wires. The three border conditions are enumerated in Figure 5.

[Border condition 1]

When the hanging pin (P1) strikes on the rim of the spherical concavity, the spherical body

cannot continue to incline as shown in Figure 5(a).

[Border condition 2]

When the wire (P₂-Q₂) is tightened under the center of spherical body, the spherical body cannot return to the standard posture as shown in Figure 5(b).

[Border condition 3]

When the two wires (P₁-Q₂ and P₁-Q₂) are located in same straight line on the surface of the spherical body, the spherical body cannot move to the direction perpendicular to the wires as shown in Figure 5(c).

Theoretical possible / impossible range for moving is calculated by taking the logical AND of the three border conditions. Upper area of Figure 6 shows the theoretical possible range for moving.

EXPERIMENTAL RESULTS OF MOVABLE RANGE

Figure 7 shows the theoretical possible / impossible range for moving and shows the experimental results of movable range. Because the motion of the spherical body is symmetrical, analyses and experiments were performed in the incline directions in which their longitudes θ are 0 to 180 degrees.

When the turn angle of ω is 0°, the maximum incline angles of $\beta_{\max}$ corresponded with the theoretically movable boundary against the incline directions which longitudes θ are from 0° to 50° and from 130° to 180° as shown in Figure 7(a).

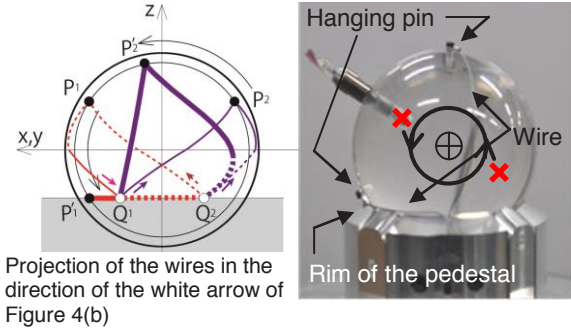
When the turn angle of ω is 30°, the maximum incline angles of $\beta_{\max}$ corresponded with the theoretically movable boundary against the incline directions which longitudes θ are from 0° to 70° and from 150° to 180° as shown in Figure 7(b).

When the turn angle of ω is 60°, the maximum incline angles of $\beta_{\max}$ corresponded with the theoretically movable boundary against the incline directions which longitudes θ are from 0° to 80° and from 160° to 180° as shown in Figure 7(c).

When the turn angle of ω is 90°, the maximum incline angles of $\beta_{\max}$ corresponded with the theoretically movable boundary against the incline directions which longitudes θ are from 0° to 180° as shown in Figure 7(d).

From the experiments, other conditions to decide the theoretical movable boundary may exist.

(a) [Border Condition 1]

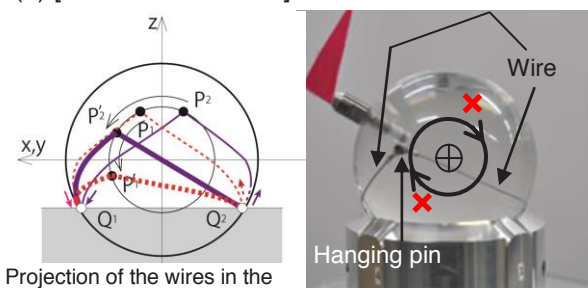


Projection of the wires in the direction of the white arrow of Figure 4(b)

[Case 1]

When the spherical body is turned with $\omega=0^\circ$ and is inclined to the direction of the longitude of $\theta=30^\circ$ and is inclined with $\beta=67.2^\circ$, the hanging pin strikes on the rim of the pedestal. The spherical body cannot continue to incline.

(b) [Border Condition 2]

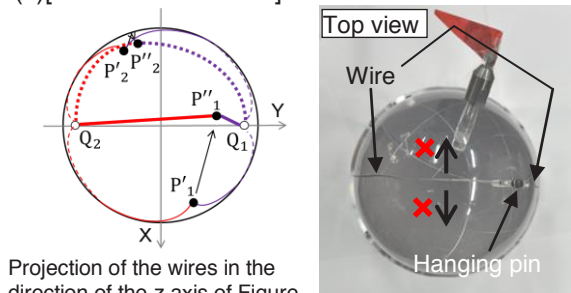


Projection of the wires in the direction of the white arrow of Figure 4(b)

[Case 2]

When the spherical body is turned with $\omega=0^\circ$ and is inclined to the direction of the longitude of $\theta=75^\circ$ and is inclined with $\beta=82.8^\circ$, the wire is tightened under the center of spherical body. The spherical body cannot return to the standard posture.

(c) [Border Condition 3]



Projection of the wires in the direction of the z axis of Figure 4(b)

[Case 3]

When the spherical body is turned with $\omega=30^\circ$ and is inclined to the direction of the longitude of $\theta=140^\circ$ and is inclined with $\beta=79.4^\circ$, the two wires are located in same straight line on the surface of the spherical body. The spherical body cannot move to the direction perpendicular to the wires.

FIGURE 5. Theoretical analysis of the spherical motor of possible / impossible range for moving

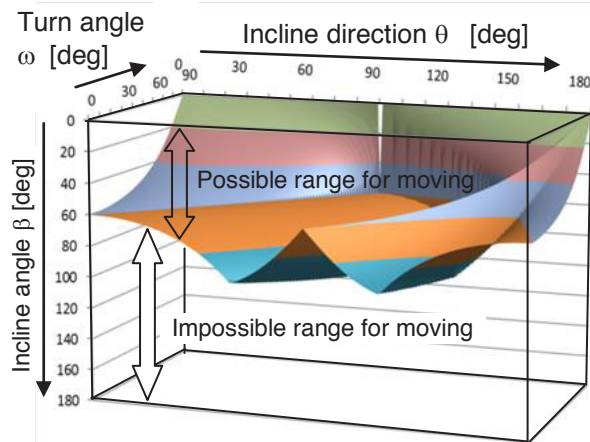
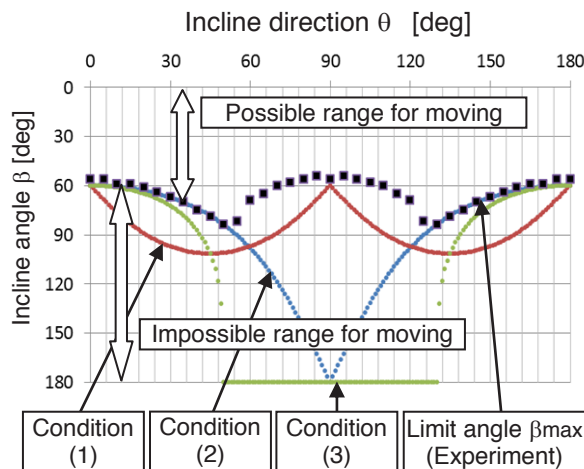
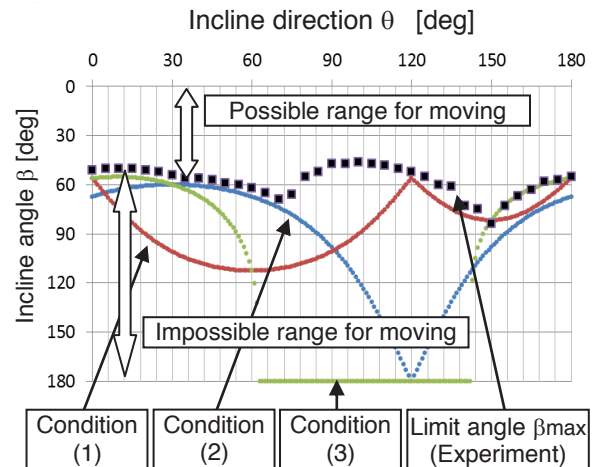


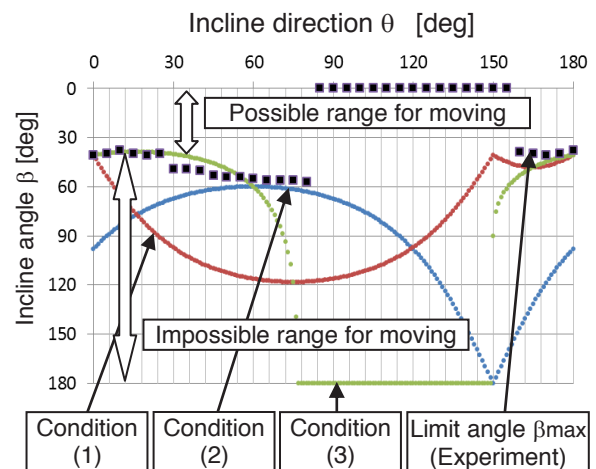
FIGURE 6. Theoretical possible / impossible range for moving



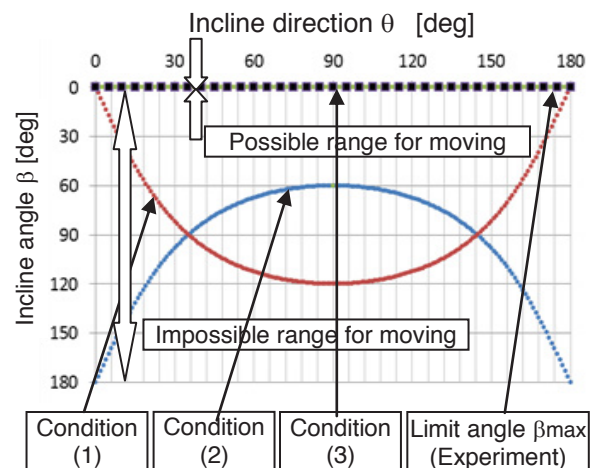
(a) Turn angle : $\omega=0^\circ$



(b) Turn angle : $\omega=30^\circ$



(c) Turn angle : $\omega=60^\circ$



(d) Turn angle : $\omega=90^\circ$

FIGURE 7. Theoretical possible / impossible range for moving and experimental results of movable range

CONCLUSIONS

A novel spherical motor manipulated by four wires is developed. Movable range of the spherical motor is theoretically analyzed and confirmed by experiments.

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TOWARD A MODELING FRAMEWORK FOR LINEAR STAGES

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INTRODUCTION

Through modeling a precision air-bearing stage, the authors demonstrate a framework for economical linear-stage model development and a frequency-weighted metric for model accuracy. Accurate modeling of a precision motion device has many advantages, including performance prediction and control development. Users can better determine what equipment will meet their performance needs without spending extra money, and manufacturers can identify gaps or overlaps in their product lines and refine their products accordingly. Furthermore, with better modeling, more refined control is possible [1], enabling better performance from existing equipment. Manufacturers can also use accurate models for model-in-the-loop (MIL) testing to develop their products more efficiently [2]. With MIL testing as part of the development process, new designs for controls and hardware can be tested virtually before resources are fully committed to a complete prototype, thereby increasing the concept-to-production efficiency.

In order to improve performance prediction and enable feedforward control of linear stages, most modeling efforts target a specific application or system aspect and evaluate the model accuracy (or model-based control performance) using a metric relevant to the targeted application [3, 4, 5, 6]. For example, settling time may be used to evaluate model-based control performance (implying model accuracy) for a pick-and-place process, whereas root-mean-squared (RMS) or peak position error may be used to evaluate the same for contouring. It can therefore be difficult to determine how to start modeling a system without a dedicated application and how to proceed in making the model more complex when higher accuracy is required.

To increase the accessibility of the benefits, the authors take the first steps in creating a modeling framework whereby sufficiently accurate models

can be produced in an efficient manner. Usually, a particular aspect of a system, e.g., friction, is targeted for study and modeling. Then the model of that particular aspect is shown to improve some performance criterion that is specific to a single class of applications, e.g., photolithography [7] or machining [8]. In contrast, the modeling framework enables the development of general purpose models that are useful for feedforward control and performance prediction. The framework is demonstrated using an Aerotech ABL1500.

EVALUATION

For validation, step responses and contour tracking experiments were conducted. The step experiments consisted of 100 trials each of four step sizes (150 mm, 5 mm, 10 μ m, and 100 nm) with sinusoidal acceleration, which was set to an amplitude of 10 000 mm/s². The steps were performed symmetrically about the center of stage travel. For example, the 150 mm move was from -75 mm to 75 mm with 0 mm at the center of travel. The contour was the one-dimensional analog of a straight line with three evenly spaced, tangential circles (5 mm, 1 mm, and 100 μ m in diameter). Beginning at the negative end of travel (-100 mm), the stage moved to -75 mm, immediately performed one period of a sinusoid with 5 mm amplitude, proceeded to 0 mm and performed a 1 mm sinusoid, proceeded to 75 mm and performed a 100 μ m sinusoid, and finally, moved to the positive end of travel (100 mm) to conclude the contour. For the 100 trials of the contour, the maximum acceleration was set high enough that the drive amplifier was operating near its maximum allowable peak current (7.5 A).

Each model was used in closed-loop simulations with the same references as the actual stage (step and contour commands), and the resulting trajectories were compared using peak and RMS position error. The move-and-settle times from the simulated step responses were also calculated and compared to those from the experimen-

tal data. Additionally, the measured position trajectories were applied to the inverse model, and the resulting current predictions were compared to the actual currents using peak and RMS error.

Finally, the frequency responses from current to position of the model and plant were compared using a frequency-domain metric. The proposed metric, η , is based on the idea that the frequency response of a perfect model should be the same as the frequency response of the actual plant, and the ratio between the two should, therefore, be unity at all frequencies. The definition of η is given as:

$$\eta = \frac{1}{1 + \left\| W(j\omega) \left(1 - \frac{\hat{P}(j\omega)}{P(j\omega)} \right) \right\|_p} \quad (1)$$

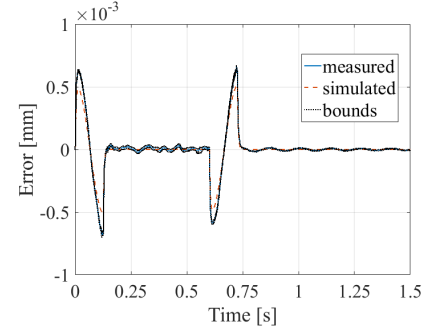
where $W(j\omega)$ is a frequency weighting function, $\hat{P}(j\omega)$ is the frequency response of a plant model, and $P(j\omega)$ is the frequency response of the actual plant. Although least-squares model evaluation is not new [9, 10], the proposed normalization by the plant frequency response function (FRF) gives the metric a definite indication of “how much” of the plant is modeled. For this work, the weighting function was chosen to be uniform, and the 2-norm was chosen for its broad base in the literature. To provide a frame of reference for this metric, a model is only beneficial for feedforward when $\eta \geq 0.5$, and $\eta = 1$ indicates a perfect model. For values less than 0.5, the model will be detrimental in the considered bandwidth when used for feedforward.

MODELS

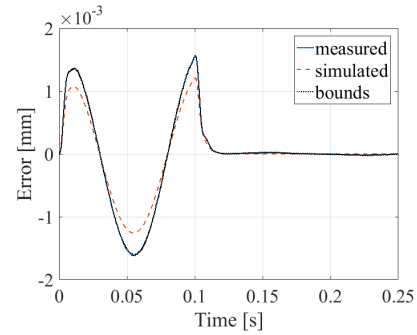
First, a simple free-mass model was defined using the nominal values of mass and force constant given in the manufacturer’s catalog. The catalog values for the force constant and mass are $k_\tau = 26.67 \text{ N/A}$ and $m = 5.1 \text{ kg}$, respectively, yielding the nominal mass model

$$\hat{P} = \frac{26.67}{5.1s^2} = \frac{5.229}{s^2} \text{ m/A} \quad (2)$$

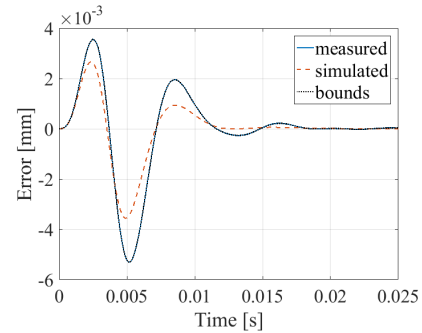
The nominal mass model was then evaluated as described in the previous section. The position error plots for the steps are shown in Figure 1. The mean (denoted “measured”) and bounds of the 100 trials are compared to the simulated response. Note that the bounds are very close to the mean. A similar plot for the contour is shown in Figure 2. The FRFs of the plant and nominal mass model are shown in Figure 3. The validation results indicated that the nominal mass



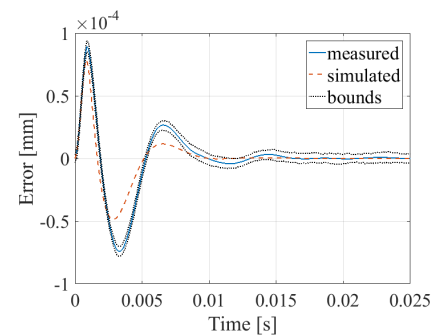
(a) 150 mm



(b) 5 mm



(c) 10 μm



(d) 100 nm

FIGURE 1. Position error for nominal mass step simulations with respect to measured data. The four step sizes (150 mm, 5 mm, 10 μm, and 100 nm) cover a wide range of amplitudes.

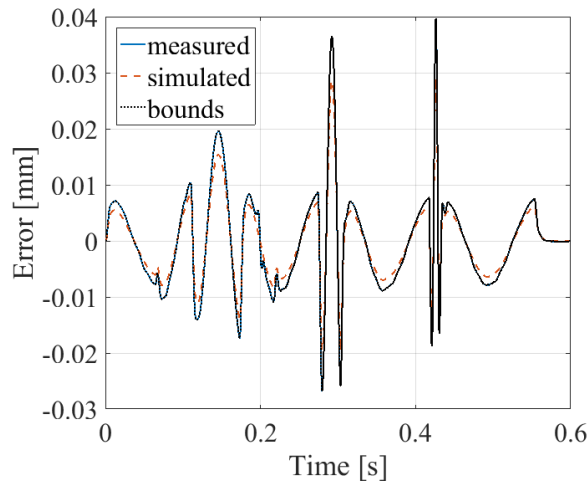


FIGURE 2. Position error for nominal mass contour simulation with respect to measured data.

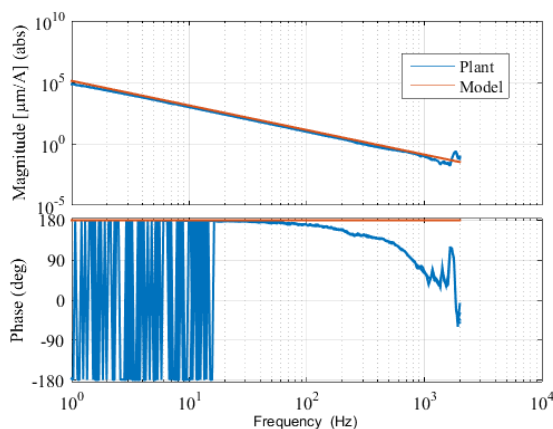


FIGURE 3. Measured and nominal mass model FRFs from current to position.

model accounted for a significant portion of the plant dynamics, but there was still room for improvement. As seen in Figure 3, a time delay is present but was left for future work as the cable drag (discussed later) was thought to be more important.

The second model increased complexity by effectively computing the mass. More precisely, the force constant-to-mass ratio was identified because the two values can be combined into a single parameter. As frequency response measurements had already been made, the identified mass model was determined by a least-squares (LS) fit to the measured plant FRF, requiring more analysis than the previous model—but no new ex-

periments. In practice, the nominal mass model would require no experiments and the LS mass model would require an identification experiment, but as the FRF measurements had already been conducted for model validation, that data was reused. Furthermore, frequency response measurements are often conducted for feedback tuning purposes, so in many cases, the data collection would be done regardless of the modeling. The LS mass model, defined as

$$\hat{P} = \frac{3.967}{s^2} \text{ m/A} \quad (3)$$

was subjected to the same evaluation methods as the nominal mass model, and the results showed that improvements in accuracy were achieved with a small increase of modeling effort. For brevity, the figures are not presented here but will be summarized along with the other results in later sections.

A spring-mass-damper (SMD) model (the third model discussed) was also fit to the FRF data using the LS criteria as the next step toward increasing model complexity and accuracy was to model the cable drag; cogging was nonexistent (the linear motor does not contain iron cores) and force ripple was small compared to the elasticity of the cabling, which included an air hose in a cable carrier and a ribbon cable. The SMD model was expected to capture the cable drag as a linear stiffness; however, the low stiffness produced a resonance at a frequency below 1 Hz and was not in the FRF data. Therefore, the resulting model did not show an improvement over the previous models when evaluated. To develop a more accurate SMD model (the fourth model discussed), it was necessary to take special care to capture data representing the low stiffness. While this could be done with FRF measurements at very low frequencies, waiting for several cycles of such low frequencies becomes prohibitively time-consuming. Instead, a constant velocity experiment was conducted to directly measure the position-dependent forces, i.e. cable drag. The average of 10 trials is shown in Figure 4. Although the drag force was hysteretic, the simplest approach was to ignore the nonlinearity and find an average linear fit. The cable drag was then added to the mass model as a linear spring, yielding

$$\hat{P} = \frac{3.8039}{s^2 + 0.1314} \text{ m/A} \quad (4)$$

and this second SMD model was evaluated,

showing some improvement as described in the next section.

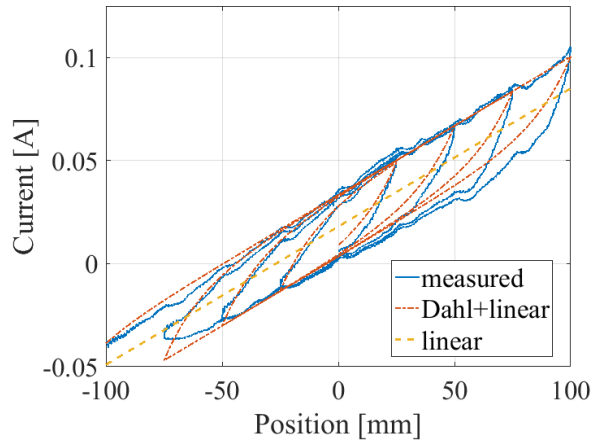


FIGURE 4. Constant velocity data with linear and linear-plus-Dahl fits. Note that cable force is proportional to current.

Next, the fifth model, with more complexity, was formed by adding a Dahl hysteresis model derived from the constant-velocity data in Figure 4. The Dahl model can be stated as

$$\frac{di_h}{dx} = \sigma \left| 1 - \frac{i_h}{i_c} \right|^\delta \operatorname{sgn} \left(1 - \frac{i_h}{i_c} \right) \quad (5)$$

where i_h is the current due to the hysteresis, x is displacement, σ is the initial stiffness (in terms of current), i_c is a constant describing the maximum value the current approaches, and δ is a shape factor. Although it was developed to describe friction [11], the Dahl model is a conceptually simple hysteresis model, making it useful for other applications. The main parameters of the Dahl hysteresis model are the initial stiffness and the saturation point, which can be visually estimated from data; the shape factor is often taken as $\delta = 1$ to simplify calculations, which was done for this work. Although parameters were fit with both genetic and pattern search algorithms, the resulting estimates (see Figure 4) differed very little from initial estimates. The parameters were $\sigma = 0.001 \text{ A/mm}$ and $i_c = 0.67 \text{ A}$. The new plant model was then reevaluated to show the improvement provided by the extra complexity.

RESULTS

The five presented models were compared side-by-side to the measured data as previously described. The relative settling times are shown in Figure 5; the various time-domain errors for

the step and contour responses are shown in Figure 6 and Figure 7, respectively; and the frequency-domain metric values are shown in Figure 8.

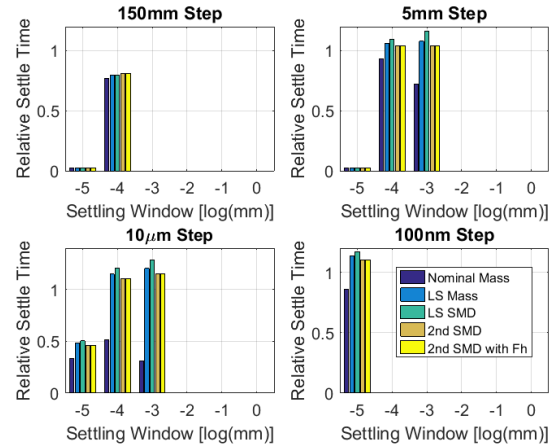


FIGURE 5. Simulated settling time normalized by measured settling time. A value of 1 is ideal.

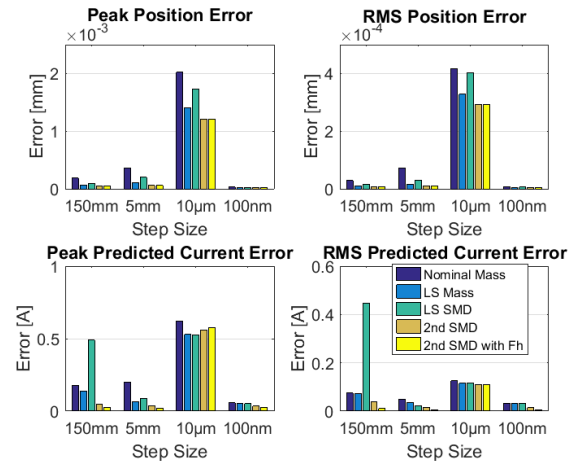


FIGURE 6. Model errors with respect to measured step response data.

DISCUSSION

The incremental increases in the model accuracy, as measured by the proposed metric as well as the conventional peak and RMS position and current prediction errors and settle times, were compared to the resources required to take each modeling step. The nominal mass model required only the catalog values and was least expensive in terms of resources. As is apparent in all of the considered metrics, the nominal mass model is somewhat helpful for predicting performance and current, but it certainly does not capture all of the plant dynamics. The nominal model performed most poorly in many cases, but the LS

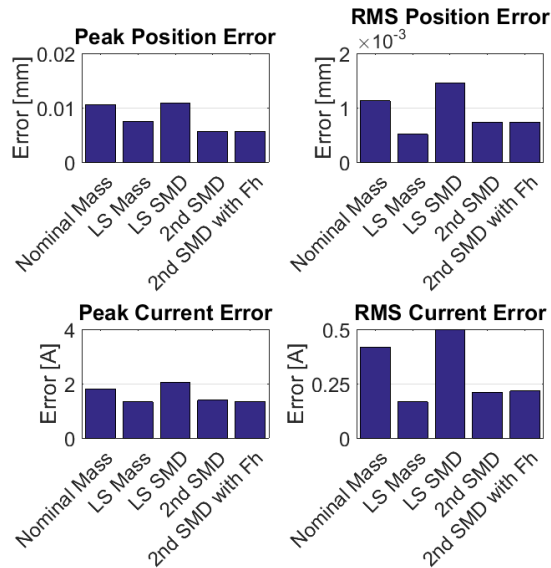


FIGURE 7. Model errors with respect to measured contour tracking data.

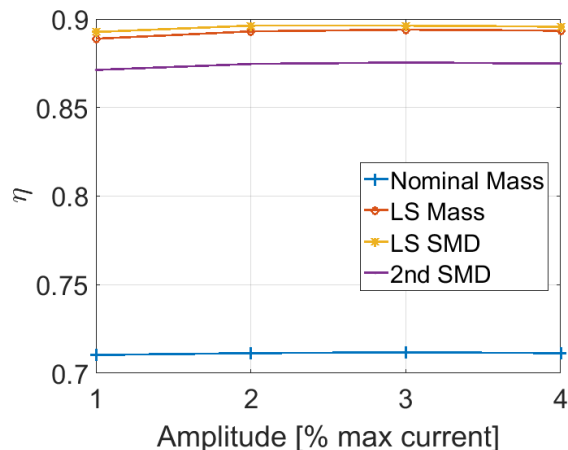


FIGURE 8. Frequency-domain metric for the 1 Hz to 100 Hz range.

SMD model (with poorly estimated stiffness) was worse with regard to the contour data.

The LS mass model accounts for a large portion of the dynamics but requires an FRF experiment; however, the FRF experiment is common practice for feedback control design and may not represent a significant increase in modeling effort. The LS algorithm itself is simple to implement and adds little to the overall modeling cost. Although the LS SMD model includes extra degrees of freedom, the fit data must capture the frequency range that includes the resonance or be noise free; if not, small perturbations in the measured

FRF may cause a poor estimation of stiffness which is detrimental to the model performance as shown in Figure 6. Therefore, this model is likely not worth the effort of measuring the very low frequencies necessary. Instead, a constant-velocity time-domain experiment is recommended as it may be captured as part of a homing cycle and adds less to the modeling cost. This second SMD model demonstrated better agreement with measured data in the step experiments than the previous models.

Finally, the hysteresis noted in the constant-velocity data was captured using a Dahl friction model. This model proved to perform very similarly to the second SMD model in the closed-loop simulations, but including the hysteresis did lead to improvement in the predicted current for the step experiments with the exception of the peak current error for the 10 μ m move. The model including hysteresis was the best predictor of settling time (see Figure 5) along with the second SMD model. For the contour experiments, the two most complex models only performed best in terms of peak position error; otherwise, the LS mass model was the best predictor of both peak and RMS current errors and RMS position error.

In terms of the frequency-domain metric shown in Figure 8, the LS SMD model appears to have the highest values, but this did not correspond to the best agreement with position in closed-loop simulation or predicted current. Because the actual stiffness of the cable creates a resonance well below the frequency range measured, fitting those dynamics amounts to extrapolation which is sensitive to even small perturbations at the lowest measured frequencies. The LS mass model scored the second best and performed best in terms of the RMS position and predicted current errors for the contour data. The frequency-domain metric was not calculated for the hysteretic model but should be quite similar to the second SMD model as they performed similarly according to most of the other metrics. The modeling costs and performances are summarized in Table 1.

CONCLUSIONS

An Aerotech ABL15020 was modeled with increasing complexity, and the models were compared with step response, contour tracking, and frequency response experiments. No single model performed best in all respects, but it was shown that the most complex model was the best

Model	Order, # Parameters	Required Data	Step Rank	Contour Rank
Nominal Mass	2, 0 (2 fixed)	None	4	4
LS Mass	2, 1	FRF	3	1
LS SMD	2, 3	FRF	5	5
2nd SMD	2, 3	FRF and CV	2	2
SMD with Dahl	2 (linear part), 6	FRF and CV	1	3

TABLE 1. Summary of modeling effort and performance. CV denotes constant velocity data.

predictor of the step response data. The contour data was best predicted by the LS mass model in terms of all but peak position error. These results indicate that a modeling framework may consider a more robust model for contouring applications but a more intricate model for well-defined, repetitive motions such as step responses. The time delay will be modeled and evaluated in future research to determine when it should be included in terms of complexity versus performance, which will also be formalized quantitatively. Further research will be conducted to determine if black-box system identification approaches (e.g., ARMAX and NARMAX modeling) are robust enough to adequately predict contour motions and if such models may outperform the grey-box SMD model with Dahl hysteresis for step motions. Finally, the research will trend toward increased formalism in evaluating complexity versus performance (drawing from fields such as system theory and process management), automated model selection and construction based on the resulting complexity/performance trade-off analysis, and inclusion of other motion systems with different components (bearings, amplifiers, motors, etc.) within the framework.

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FEEDFORWARD ELEMENT DESIGN FOR ULTRAHIGH ACCELERATION LINEAR DRIVE MECHANISM USING LEARNING CONTROL ELEMENT

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INTRODUCTION

Linear motion systems are key components in precision industrial machines and their motion characteristics greatly influence the machine performances. High tracking accuracy and high responsiveness are important characteristics for the machines such as machine tools, assembly apparatuses. The responsiveness is often discussed using the acceleration and the velocity as important indicators of the responsiveness. Their required levels are becoming strict increasingly [1][2].

In order to realize precision and high speed motion with an ultrahigh speed linear drive mechanism, the ultrahigh-acceleration linear drive mechanisms have been designed and the driving performances of the prototypes have been examined [3]. The prototypes have shown the acceleration higher than 70G and the velocity higher than 10m/s. However they have significant nonlinear characteristics and it is difficult to use exact dynamic model in the controller design for high speed and precision motion control. The controller with a learning control (LC) element as a feedforward element has been designed and the effectiveness of the controller on the tracking error reduction has been validated experimentally [4]. However the controller always requires the learning process for each tracking motion. The requirement reduces the usability of the control system. In [4], a disturbance observer was not used for high motion accuracy although it is generally effective to reduce unknown disturbance effects removed. This paper describes the feedforward (FF) element design for precision and high speed tracking control using the LC element only in determination process of the FF element but not under motion control. The usefulness of the controller with the designed FF element and a

disturbance observer was experimentally validated.

ULTRAHIGH-ACCELERATION LINEAR DRIVE MECHANISM

Figure 1 shows a prototype of ultrahigh-acceleration linear drive mechanism. This linear drive mechanism includes a moving type permanent magnet linear synchronous motor (MPM LSM). The weight of the mover are 4.63kg. The mover is supported by a linear ball guide. The stator has two electromagnet lines which are located on both sides of the mover. The displacement of the mover is measured using a laser position transducer (Resolution: 79.1 nm). This mechanism can generate the larger thrust force than 3000N and show the higher acceleration than 79 G. However it has significant nonlinear characteristics resulting from nonlinear static thrust gain, friction characteristics, the thrust ripple.



Fig. 1 Prototype of ultrahigh-acceleration and high-velocity linear drive mechanism

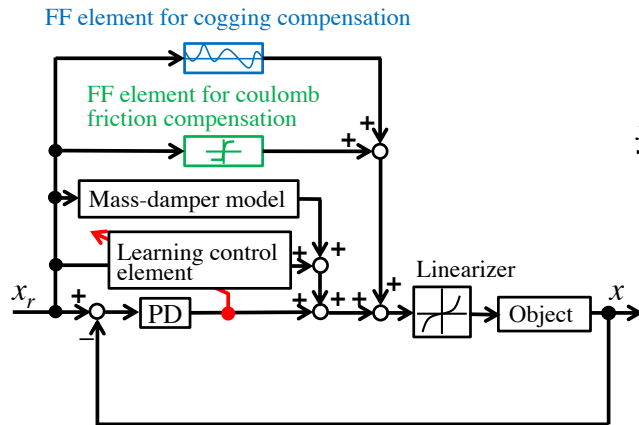


Fig. 2 Control system with an on-line learning control element

PREVIOUS CONTROL SYSTEM

Figure 2 shows the control system with a learning controller for the MPM LSM. The controller has a two-degree-of-freedom (2DOF) compensator consisting a PD element and a FF element. The FF element is designed based on the simple dynamic model for compensating the mover mass and damping effects. PD gains are adjusted for reducing errors. For suppression of the nonlinear effects, two FF elements for the cogging and friction force compensation and a linearizer are integrated into the system.

The learning controller (LC) works for suppressing residual reproducible errors.

The significant nonlinear characteristics of the MPM LSM makes it impossible to construct an exact dynamic model for the FF design. The LC element can be designed without the exact dynamic model and can provide high speed precision tracking performances although the controller always requires the learning process for each tracking motion. Disturbance observers are generally effective to reduce bad influences of unknown disturbance factors in low frequency region. However the disturbance observer is not used in Fig.2 for high motion accuracy [4].

FF ELEMENT DESIGN CONCEPT USING THE LC ELEMENT

The previous control system has three invariant FF elements which have designed for different error factors individually. However they are determined as functions of a reference signal. The tracking errors are considered to include components depending on position, velocity, acceleration/deceleration motion. The effect of

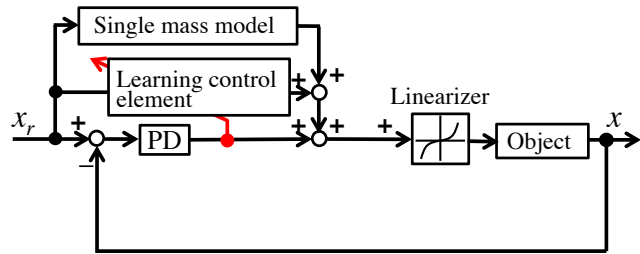


Fig. 3 Initial control system modified for determining the invariant FF elements

each parameter on the tracking error is influenced by input signals to the control system. When the target value for constant velocity motion is applied to the control system, the LC element generates the output signal for compensating the error depending on the position under the constant velocity. Thus the FF element as the function of the position and the velocity was determined from the output signals generated by different constant velocity motions. The designed FF element is expected to be useful for compensating the error depending on the position and the velocity. The FF element does not have a learning function and can be used with a disturbance observer.

FF ELEMENT DESIGN PROCEDURE

The table mass is a typical error factor excited by acceleration motion. However its value is often known and the effect is easy to compensate. Thus the simplest single mass model is used as an initial FF element. The other FF elements are designed with the LC for compensating the error factors depending on the position, the velocity and the current command to the driver amplifier. The input current command is dealt with a parameter for the thrust gain error compensation.

Figure 3 shows the initial control system modified for determining the invariant FF elements. The FF element based on the simplest single mass model is implemented as an invariant one. The FF element design procedure consists of two steps as follows.

(1) The FF element determination for the error factors depending on the velocity and the position with the current command The FF elements are determined from the constant velocity motions of the control system shown in Fig.3. Figure 4 shows the output of the LC element in Fig.3 against three constant velocity

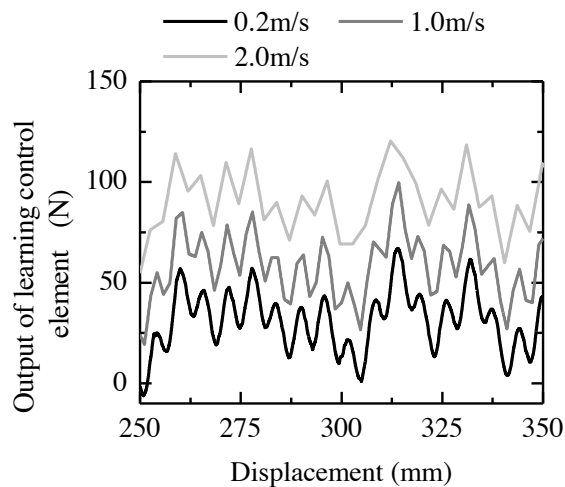


Fig. 4 Relationship between the mover displacement and the LC element output

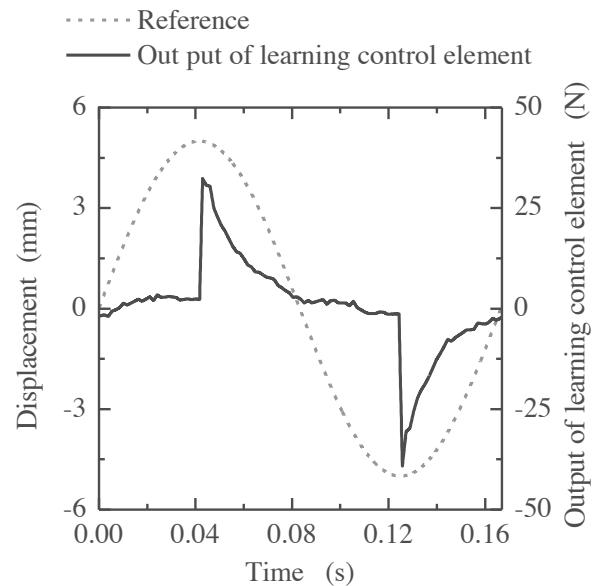


Fig. 6 Sinusoidal response of the control system in Fig.5

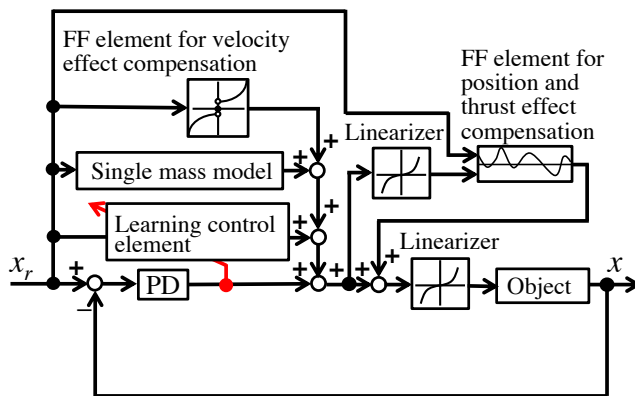


Fig. 5 Control system with the LC element and the invariant FF elements as functions of the position, the velocity and the input current command

references. In Fig.4, variant components in the outputs indicate the outputs depending on the position and the current command. The offset components indicate the outputs depending on the velocity. The outputs can be divided into the two components by a low pass filter.

(2) The FF element determination for a remaining error factor Figure 5 shows the control system comprising the control system in Fig.3 and the FF element determined in the

previous step. The LC output in the control system indicates the signal resulting from the remaining error factor. The sinusoidal response of the control system is shown in Fig.6. The output response shows the significant behavior when the motion direction changes. This behavior is independent of frequency and amplitude of input signal. This is considered to result from friction characteristics except viscosity effect [5]. In order to compensate this behavior, a new FF element is added.

TRACKING CONTROL EXPERIMENTS

Figure 7 shows the control system including the FF elements designed by the design procedure mentioned above. In order to suppress a unknown disturbance force and characteristic change, the control system has the disturbance observer designed based on the simplest single mass model. It is easy to design all the FF element and the disturbance observer without the exact dynamic model of the MPM LSM. The other control elements also can be determined without the exact dynamic model.

In order to examine the usefulness of the design procedure, the responses of the determined control system were measured with the sinusoidal input. The three tracking errors of the prototype MPM LSM with the sinusoidal input are shown in Fig.8. The light gray dotted line

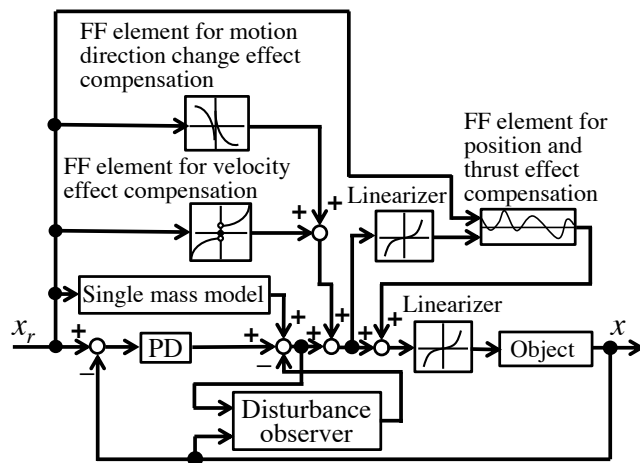


Fig. 7 Control system including the FF elements determined by the design procedure

and thin solid line indicate the results with the disturbance observer, but not the LC control element. The comparison result indicates that the designed FF elements are effective. The tracking error by the controller in Fig.7 is smaller than that by using the previous controller with the disturbance observer and is bit larger than the previous controller with the LC. However the difference between the maximum errors by the current controller and the previous controller with the LC element is small. This indicates the usefulness of the design procedure.

CONCLUSION

In this paper, the feedforward design concept and procedure for high speed and precision motion have been proposed. The design does not require an exact dynamic model and learning action during motion control. The usefulness of the design procedure was experimentally validated.

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- Reference
- Tracking error of the control system of Fig.7
- Tracking error (with disturbance observer [4])
- Tracking error (with learning controller [4])

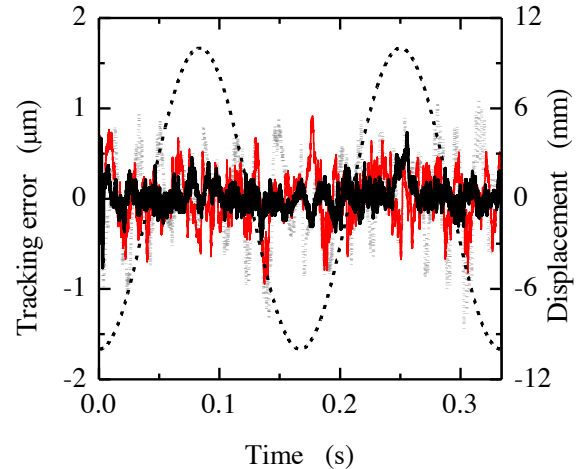


Fig. 8 Sinusoidal response of the control system in Fig.7

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MICRO-MANIPULATOR SYSTEM WITH WIRELESS ACTUATORS USING LASERS AND THERMO-SENSITIVE MAGNETIC MATERIAL - SYSTEM CONCEPT AND GRIPPING UNIT DESIGN -

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INTRODUCTION

In recent years, various applications in the field of biotechnology such as artificial insemination [1], bio-devices and assembly of biological body [2] have become more important research topics. In addition to average characteristics of cells and biological bodies as the aggregate, characteristics of each cell and each body need to be examined [3]. In the examination, suitable micro-manipulators are needed for sorting of desired cells and biological bodies, their property inspection, gripping, processing, etc. Nowadays many micro mechatronics devices have been developed and implemented to practical products for system minimization and performance improvement. There are many useful fabrication methods for micro-size objects made from various materials and shapes. Integration of biological bodies into micro mechatronics are promising research topics. For assembling various micro-size objects mentioned above, it is important to realize multi-use micro-manipulators which can operate cells and micro mechatronics devices.

Up to date, several types of manipulation systems have been reported for operating micro-size objects. Mechanical manipulators are the most widely used with microscopes. They can be used for general-purpose. However they generally need large space and many actuators which need a number of wirings. These limit number of gripping units for cooperative operation.

Optical tweezers systems can provide the function for operating many micro-size objects simultaneously. However the types of operable micro-size objects are limited and the generated force is on the order of pN [4].

The purpose of this research is to realize a micro-manipulator system with the following features;

- (1) It can operate various types of micro-size objects in air, liquid and boxed-in space.
- (2) It is easy to replace the end-effectors of the grippers for multi-purpose use.
- (3) The micro-grippers are small enough to keep observing the motions of many driving grippers under a microscope.
- (4) The driving systems are isolated from the work space and free from the influence of the environment in the space and the wiring problems.
- (5) The grippers can provide sufficient force (> 1 mN)

For these goals, wireless micro-actuators using laser beams, permanent magnets (PMs) and thermos-sensitive magnetic material (TSMM) parts are used. Since the actuators can be driven by laser beams, they have simple structures and free from wiring.

This paper describes the concept of the micro-manipulator system and explains the design and the basic characteristics of the prototype gripping unit.

CONCEPT OF MICRO-MANIPULATOR SYSTEM WITH WIRELESS ACTUATORS Driving Principle of Wireless Actuator

The saturation magnetic flux density and the magnetic permeability of the TSMM decrease with the increase of temperature. The characteristics can be observed even if the temperature is not so high. Figure 1 shows the driving principle of the wireless actuator with the TSMM. The two PM pairs produce magnetic flux which passes through the TSMM. When a laser beam irradiates and heats the high magnetic flux density area on the TSMM as shown in Fig.1(b), the magnetic resistance on the area increases. Thus the force is applied to the TSMM for moving the hot area out of the magnetic path

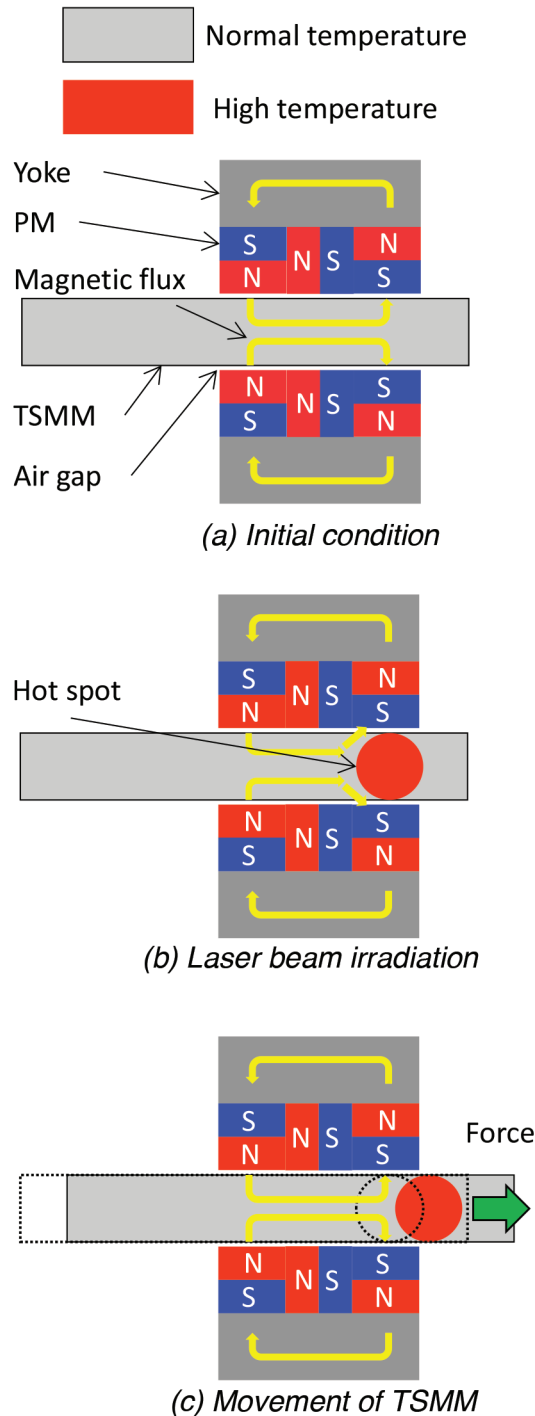


Fig. 1 Driving principle of the actuator with thermo-sensitive magnetic material

and reducing the magnetic resistance (see Fig.1(c)). When the laser spot is fixed on the TSM to the PM pairs, the force is applied

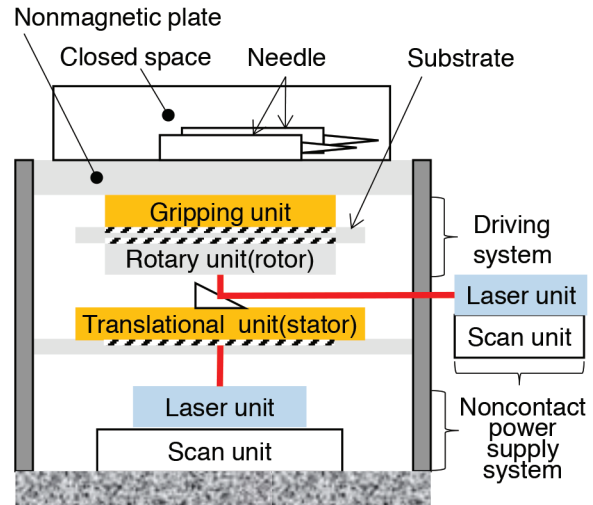


Fig. 2 Schematic of a micro-manipulator with a single gripper

continuously. The moving direction can be switched by the change of hot spot position on the TSM. Thus the actuator can be controlled without wiring.

Basic System Configuration

Figure 2 shows an example of basic system configuration for the manipulator system. The driving system is isolated from the work space by nonmagnetic plate and free from the environment of the work space. The driving system is coupled to the needles via magnetic coupling. The micro-size objects can be operated with the driving systems when the needles are in the work space.

The driving system consists of three function units such as gripping, rotary and translational units. They are stacked as shown in Fig.2. All the units are driven by two laser beams. The lower laser unit is used for translational motion and the side laser unit is for gripping and rotary motion. Their function units have several spot area to be irradiated by a laser beam and a valid function depends on heating spot. Thus the substrate between the units is transparent and the spot areas are not overlapped with each other in the vertical direction.

PROTOTYPE OF GRIPPING UNIT

Figure 3 illustrates the configuration of the prototype of the gripping unit. Two needles are isolated from the gripping unit by the nonmagnetic plate and connected via magnetic

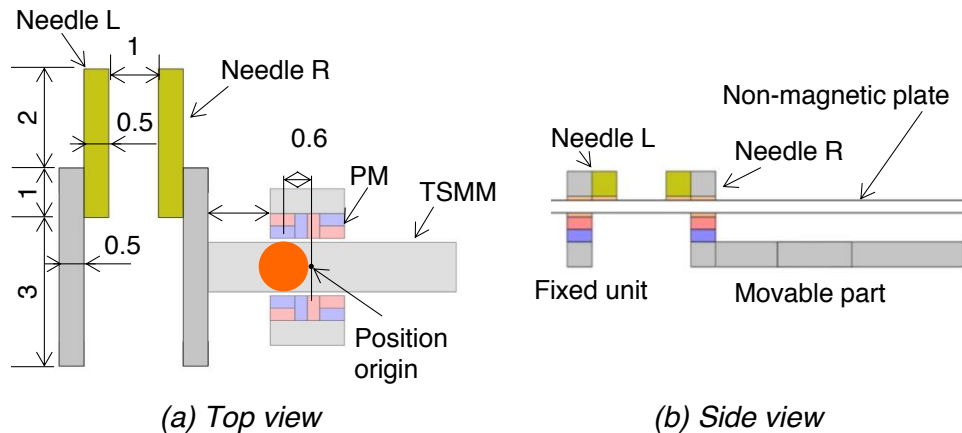


Fig. 3 Configuration of the gripping unit

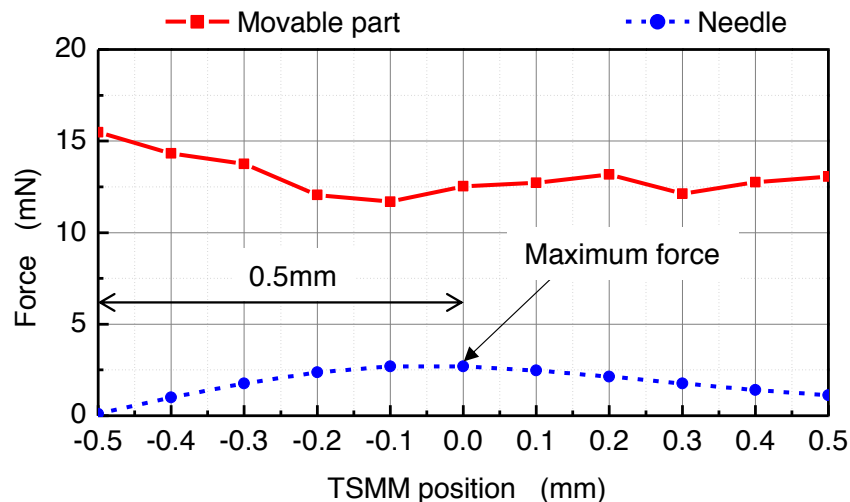


Fig. 4 Generated force of the actuator obtained by using the magnetic analysis

coupling. The right side needle is driven by the wireless actuator through magnetic coupling. The installing area is smaller than 10 mm x 10 mm.

Figure 4 show the generated force of the actuator obtained by using the magnetic analysis. In the analysis, the initial temperature of the TSM part is 22 degree Celsius. The hot area is in a 1mm diameter circle. Its temperature is 80 degree Celsius. The generated force depends on the hot spot position. This result indicates that the actuator can produce the force larger than 10 mN. However the magnetic coupling makes the maximum gripping force 2.7 mN.

Figure 5 shows the overview of the experimental setup. The actuator is irradiated with the laser beam emitted from the light source through the reflector from below. The motion of the actuator is observed by the camera. The spot size and the output power of the laser beam are 1 mm and 1 W, respectively.

Figure 6 shows the prototype of the gripping unit using the actuator. The sliding guide parts are attached for straight motion of the movable part on the glass plate with magnet pairs. The glass plate has an antireflection layer on it. When the laser beam was irradiated on the TSM, the displacement of the needle was larger than 0.6 mm for 0.4 s.

CONCLUSION

In this paper, the concept of micro-manipulator system using wireless actuators with the TSMM has been introduced and the design and basic characteristics of the prototype gripping unit have been shown.

ACKNOWLEDGEMENT

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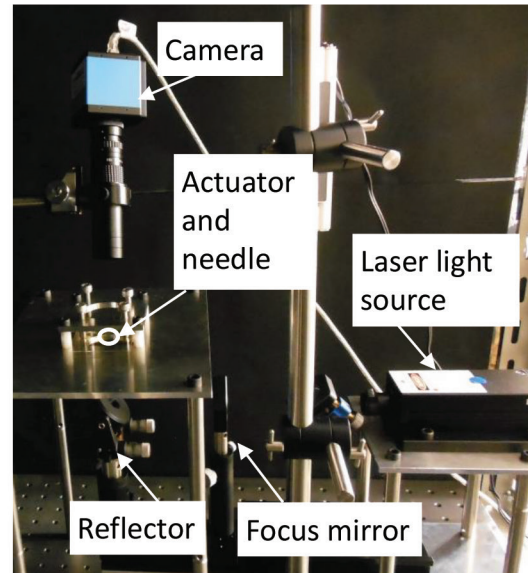


Fig. 5 Experimental setup

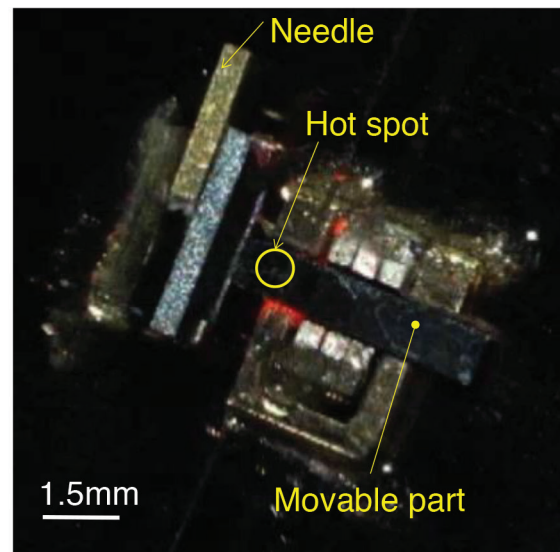


Fig. 6 Prototype of the gripping unit using the wireless actuator

Measurement of vertical displacement of water driven stage for pitching control during feed motion

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INTRODUCTION

Ultra-precision machine tools are used for the machining of various precision parts. In recent years, downsizing of the parts and improvement of the part accuracy are strongly desired. Thus, the improvement machining accuracy of the ultra-precision machine tools is required as well. The motion accuracy of the machine tools influences the machining accuracy directly. Therefore, improvement of the motion accuracy of the feed table of the machine tools must be improved so that the required machining accuracy is achieved. Thus, the water driven stage was developed as the feed table of ultra-precision machine tools in our previous study [1].

The table of the stage is supported by water hydrostatic bearings. In general, the lubricating fluid of the hydrostatic bearings is oil or air. However, if water is used as the lubricating fluid, friction heat is suppressed compared to oil. The viscosity of water is smaller than that of oil. In addition, water is an incompressible fluid unlike air, as a result water hydrostatic bearings is advantageous for designing high bearing stiffness and load capacity. Furthermore, the water driven stage is expected to be high thermal stability, because water has the high heat capacity and the thermal conductivity.

The motion error of the feed table includes error of the table position and the table attitude such as pitching. In our previous study [2], the pitching control system of the water driven stage was designed and tested experimentally. The control tests however indicated that the pitching control changes undesirable vertical displacement that is normal direction of the feed direction of the table. The vertical displacement of the table of the water driven stage can be controlled by controlling the pressure balance of the water hydrostatic bearings. However, the vertical

displacement of the table has to be measured in order to control.

However, it should be noted that the measurement error of the vertical displacement of the table is occurred due to various factors such as the form error of the measuring object. In addition, the pitching motion affects the measurement of the vertical displacement of the table. The purpose of this study is to propose and investigate a measurement method of the vertical displacement of the center of gravity of the table during feed motion.

WATER DRIVEN STAGE

Figure 1 shows the structure of the water driven stage. The table of the stage is driven by supplying water into the internal cylinder piston mechanism. The table of the water driven stage is supported by water hydrostatic bearings without direct contact to the guideway. The water hydrostatic bearings of the stage are composed of 12 bearing recesses. In addition, the water driven stage has independent water supply ports to each recess. Therefore, the table attitude such as pitching can be controlled by changing pressure balance of the bearings that can be controlled by the flow rate of the supply water into each bearing.

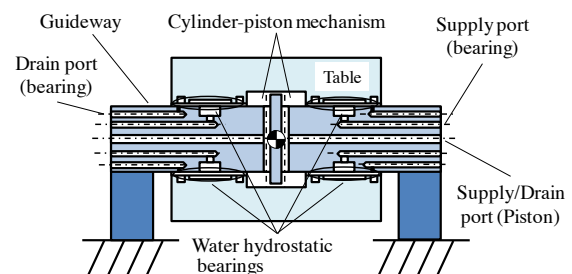


FIGURE 1. Structure of water driven stage

NOMENCLATURE

θ	:pitch angle of table
θ_d	:desired pitch angle of table
Z	:vertical displacement of center of gravity of table
Z_m	:measured vertical displacement table
Z_e	:measurement error of vertical displacement of table due to form error of straight edge
ΔZ_m	:difference of output of two capacitance sensors
x	:displacement of center of gravity of table in x-axis
x_m	:output of displacement sensor
x_s	:distance between two capacitance sensors
h	:distance between the center of gravity of the table and the top surface of straight edge
e_θ	:steady state error of pitch angle of table
$P_{r1,2,3,4}$	:recess pressure of water hydrostatic bearings at B ₁ , B ₂ , B ₃ , and B ₄ .
$\Delta P_{r1,2,3,4}$	:change of recess pressure of water hydrostatic bearings at B ₁ , B ₂ , B ₃ , and B ₄ .

PRINCIPLE OF PITCHING AND VERTICAL DISPLACEMENT CONTROL

Figure 2 shows the principle of the pitching control of the table of the water driven stage. As shown in Fig. 2, the table can be rotated counterclockwise by increasing the recess pressure of the bearings at B₁ and B₄ and decreasing the recess pressure of the bearings at B₂ and B₃. In addition, the table is moved upward by increasing the recess pressure of the bearings at B₁ and B₂ and decreasing the recess pressure of the bearings at B₃ and B₄. Similarly, the table can be rotated clockwise and moved downward by changing recess pressure of each bearing. Furthermore, the recess pressure can be controlled by the water flow rate into each water hydrostatic bearing.

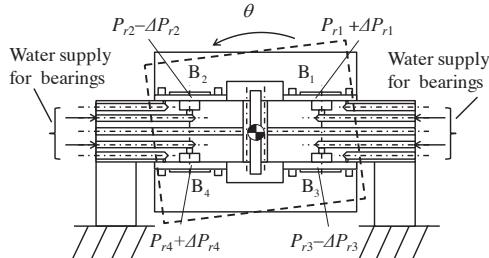


FIGURE 2. Principle of pitching control for water driven stage

NECESSITY OF VERTICAL DISPLACEMENT CONTROL FOR PITCHING CONTROL

Figure 3 shows experimental rig for the pitching and vertical displacement control of the water driven stage. In order to control the pitching and vertical displacement of the table, the recess pressure is controlled by controlling the flow rate of the supply water into each bearing. The flow rate can be controlled by electrically controllable flow control valves that are attached to the flow channels of the bearings B₁ and B₂ as depicted in Fig. 3. Our previous study revealed pitching error of about 0.5×10^{-3} degrees of the water driven stage during feed motion [2].

As shown in Fig. 4, the pitching error of 0.5×10^{-3} degrees makes error of the depth of cut about $0.13 \mu\text{m}$ per 15 mm during the planing process, for example. The error of the depth of cut degrades machining accuracy of the ultra-precision machine tool. Therefore, in order to improve machining accuracy, the table pitching of the stage is needed to be controlled.

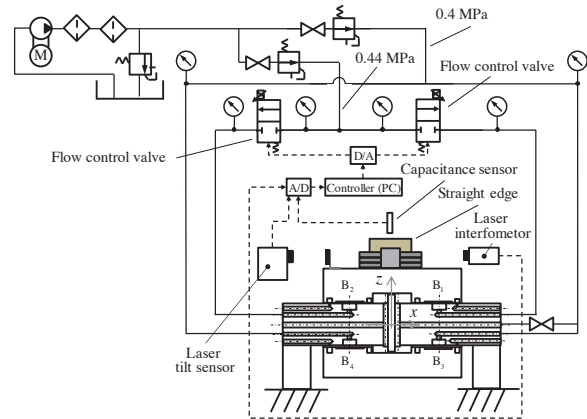


FIGURE 3. Pitching control system with open-loop vertical displacement control system

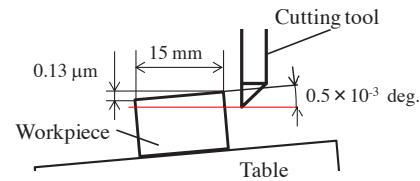


FIGURE 4. Influence of pitching error of table on depth of cut

In our previous study, the pitching control system was designed and tested. As shown in Fig. 5(a), the control system can control the pitching of the table. However, as presented in Fig. 5(b), the pitching control changes not only pitching but also undesirable vertical motion of the table that makes error of the depth of cut as given in Fig. 4.

Therefore, it is needed to integrate the vertical displacement control system into the designed pitching control system.

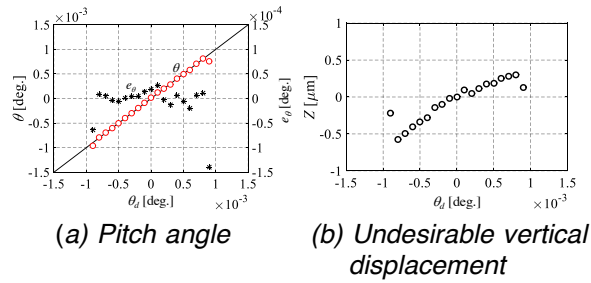


FIGURE 5. Static characteristic of pitching control without vertical displacement control

MEASUREMENT METHOD OF VERTICAL DISPLACEMENT OF CENTER OF GRAVITY OF TABLE

Measurement Principle

In order to control the vertical displacement of the center of gravity of the table, the vertical displacement is needed to measure precisely. However, in the case of the table moves along x-axis, namely the feed direction, the measurement error is included in the vertical motion measurement due to the pitching of the table and the form error of the measurement target as given in Fig. 6.

The vertical displacement of the center of gravity of the table can be introduced by compensating the influences of the pitching and the form error as presented in Eq. (1).

$$Z = Z_m - \left(x - h \tan \frac{\theta}{2} \right) \tan \theta - Z_e(x) \quad (1)$$

In Eq. (1), Z_m is the direct measurement data by a capacitance sensor and θ is the measured pitch angle. Furthermore, Z_e indicates the form error of the measurement target.

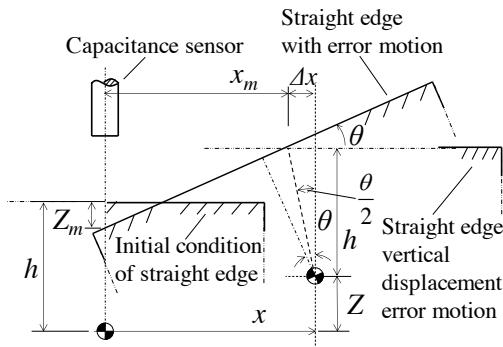


FIGURE 6. Measurement of vertical displacement of table

Measurement of Straight Edge

In order to remove the measurement error caused by the form error of the straight edge represented by Z_e in Eq. (1), the form of the straight edge was measured.

The surface of the target was measured by a 3D optical surface profiler. Figure 7 shows the measurement result of the straight edge used as the measurement target. During actual control of the pitching and vertical displacement of the water driven stage, a capacitance sensor measures averaged area of the target as shown in Fig. 8. Then form error Z_e in Eq. (1) was identified by the averaged data by taken into account of the spot size of the capacitance sensor. Obtained form error Z_e is shown in Fig. 9. It is indicated that the form error of the target is 8 nm at a maximum. An influence of the form error on the measurement of the vertical displacement is quite small however the form error Z_e is compensated according to Eq. (1).

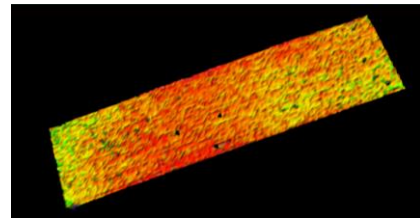


FIGURE 7. Measurement result of straight edge

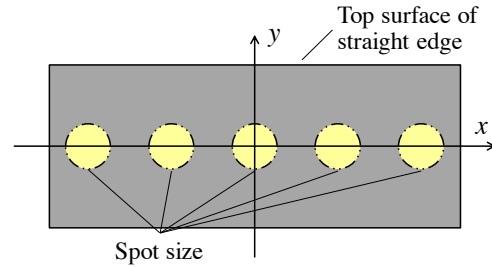


FIGURE 8. Measurement point of the straight edge

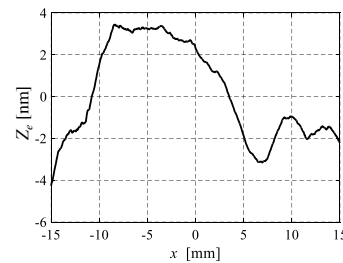


FIGURE 9. Form accuracy of straight edge

MEASUREMENT OF VERTICAL DISPLACEMENT DURING FEED MOTION

Figure 10 shows the experimental setup that was used to verify the prepared measurement method. As shown in Fig.10, the measurements were carried out during feed motion of the table with the external load of 1 kg that was applied at the off-center position of 60 mm from the center of gravity of the table. In addition, the experiment was conducted with two capacitance sensors and a laser tilt sensor.

Figure 11 shows the experimental result. The pitching of the table measured by the tilt sensor is depicted in Fig. 11(a). Direct data measured by the capacitance sensors at Z_1 and Z_2 are presented in Fig. 11(b), namely the influence of the pitching is included in the measurements result.

The objective of the measurement is to determine the vertical displacement of the center of gravity of the table. However, it is clearly verified that the displacement of the table measured at Z_1 and Z_2 are considerably different because of the pitching effect.

The compensation based on Eq. (1) was then applied to the data presented in Fig. 11(b). Figure 12 shows the compensated result. As shown in Fig. 12, the difference between the original data presented in Fig. 11(b) becomes small.

Influence of the external loads acting on the table was investigated experimentally. The experiments were carried out by putting various masses on an off-center position on the table. No feed motion was made during the experiments. Figure 13 shows the experimental result. As shown in Fig. 13, the compensation effect of the proposed method is clearly identified though further improvement is still needed. Then the pitch measurements were compared between the tilt sensor and two capacitance sensors.

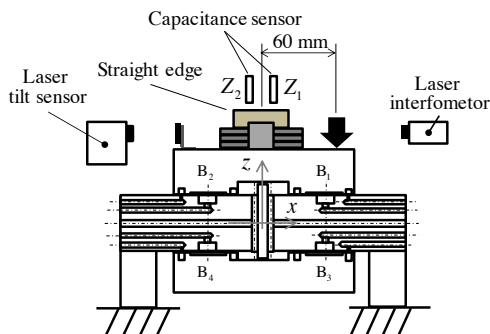
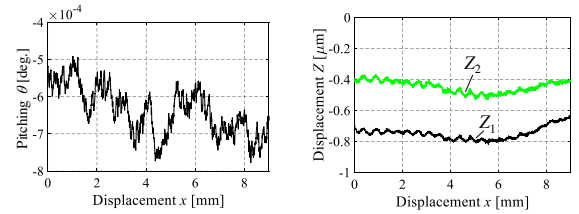


FIGURE 10. Experimental rig for measurement vertical displacement



(a) Pitch angle (b) Vertical displacement
FIGURE 11 Measured pitching and displacement

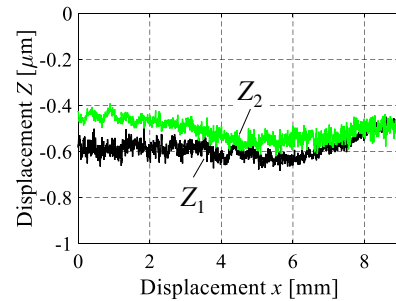


FIGURE 12 Measured vertical displacement of center of gravity

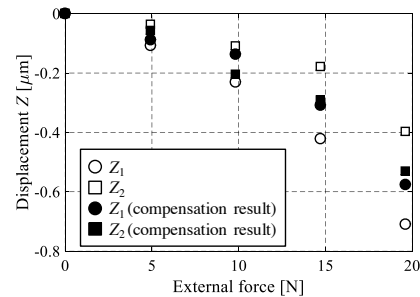


FIGURE 13 Influence of external loads on measurement of vertical displacement

As shown in Fig. 10, the vertical displacement given in Fig. 13 was measured by two capacitance sensors. Table pitching can be estimated as Eq. (2) by the distance of two capacitance sensors and the difference of the outputs of the two capacitance sensors. Measured table pitching due to the tilt sensor method and the capacitance sensor method are compared in Fig. 14.

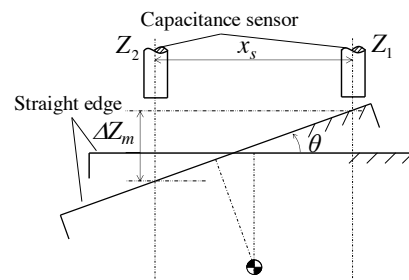


FIGURE 14 Measurement method of pitching by capacitance sensors

$$\theta = \tan^{-1} \left(\frac{\Delta Z_m}{x_s} \right) \quad (2)$$

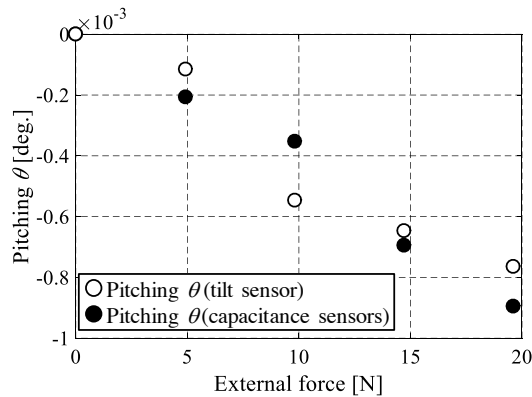


FIGURE 15 Measured pitching by tilt sensor and capacitance sensors

As depicted in Fig. 15, the pitching of table changes depending on the external loads. However, the measurement results are deferent depending on the measurement methods. Influence of the measurement of pitching on the vertical displacement of the center of gravity will be investigates in our future work.

CONCLUSION

This study dealt with the measurement of the vertical displacement of the center of gravity of the table during feed motion. The measurement error of the vertical displacement during feed motion was discussed. The compensation method that reduces the undesirable influence of pitching on the measurement of vertical displacement was considered. In addition, the form error of the measured target was measured and was compensated. Furthermore, the validity of the compensation method for vertical displacement of the table was verified via experimental studies.

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ANALOG DROP VOLUME CONTROL ENABLED BY AUTOMATED WAVEFORM GENERATION FOR PIEZOELECTRIC INKJETS

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INTRODUCTION

Drop on demand piezoelectric inkjetting is an important technology for direct patterning and film deposition in a variety of applications involving micro and nanofabrication. Advances in inkjetting are especially important for printed electronics which use inkjets for high throughput selective deposition over large areas, and for Jet and Flash Imprint Lithography [1] which uses adaptive material deposition for patterning templates with pattern density variations. Piezoelectric inkjet heads typically operate with discrete drop volume control as a multiple of the minimum volume that can be reliably jetted for a given material and a given inkjet head. Enhancing inkjet operation to include analog drop volume control would enable a much finer level of process control in the above stated applications. Today this level of precision process control is typically achieved by pushing the limits of drop volume resolution, which can lead to compromised drop placement precision, and/or reliability. A key challenge in precision volume control in piezo inkjets is the need for manual waveform tuning. Manual tuning is required due to lack of reliable reduced-order models for the fluid-structure interaction in piezo inkjets that can be used to create automated control schemes.

In this paper, we explore analog precision in drop volume control in a range where the inkjet can perform reliably with accurate drop placement. A related approach of grayscale printing technology relies on jetting multiple drops that merge before reaching the substrate [2-4]. This paradigm of digitally modulated grayscale printing, however, precludes fractional drop volumes from being delivered to the substrate to enable analog drop volume control.

To overcome this, we propose a stochastic machine-in-the loop control system for piezoelectric inkjets that enables precision analog drop volume control. The ability to specify fractional drop sizes while avoiding

manual waveform tuning has the potential to enable highly optimized material deposition. The proposed technology automatically optimizes piezoelectric inkjet actuation waveforms to achieve specified fractional drop volumes across different materials, and inkjet hardware. It also streamlines the introduction of new materials where end-use application is prioritized over the need to match rheology for inkjetting performance.

EXPERIMENTAL SETUP

A brief description of the stochastic machine-in-the-loop inkjet control system is provided next. A computer vision system monitors the behavior of inkjetted drops in real time as the inkjet is actuated by various waveforms. These waveforms are modified by a genetic algorithm where selection probability is based on the error between the target volume and the actual drop volume combined with measurement noise and the incorporation of a soft inequality constraint on drop velocity. The need for the velocity constraint is described in more detail later. Figure 1 shows the block diagram for the control system.

The system processes queues of waveforms measuring the performance achieved by actuating the inkjet with each waveform. Images of in-flight drops are captured via single-flash single-exposure stroboscopic imaging [5]. To capture an image of an inkjetted drop in flight, an LED is briefly illuminated at a set time delay in relation to the start of actuation, typically between 0 and 500 μ s. A CCD camera equipped with microscope optics captures images with an exposure time such that exactly one LED strobe is captured during each image. The frame rate is similarly matched as a multiple of the jetting frequency so that the illumination in consecutively captured images remains stable. A series of images is captured to characterize the behavior of the inkjet when actuated by a particular waveform by incrementing the strobe delay time between each image capture.

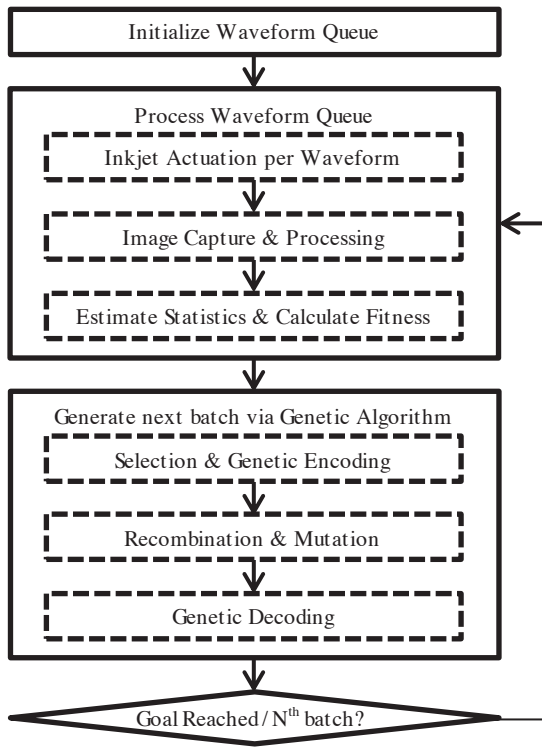


FIGURE 1. Block diagram of genetic algorithm based inkjet tuning system.

Images are processed to extract drop silhouettes and derive drop property estimates. Drop volume is estimated by disc integration, where each row of pixels is rotated about a vertical axis [6]. Instantaneous drop positions are estimated by calculating the centroid of volume using the same method as described for the drop volume estimate. The centroid of volume is the best estimate for position as it is the least affected by the contraction and oscillations of fluid ligaments in flight.

Drop volumes and velocities are estimated from the resulting time series by robust least squares where outliers are filtered using a bisquare weighting function. The drop velocity is of particular importance because drop placement accuracy scales with the reciprocal of velocity. Therefore, a minimum velocity is required to ensure adequate drop placement. Furthermore, the placement of any drops measured to travel at near-zero velocities is essentially random and is therefore considered to be a severe fault.

The genetic algorithm requires a scalar measure of performance, or fitness, to be derived for each

waveform. We use a combination of volume error in relation to a target volume, a soft inequality constraint on drop velocity, and the variance of the drop volume estimate. The drop volume variance captures drop-to-drop volume instability in addition to filtering false positives where unstable jetting surreptitiously generates a mean volume estimate that happens to be close to the target volume. Fitness is calculated by an expression of the form:

$$\frac{1}{A_1(\mathbb{V}_1 - \mathbb{V}_t)^2 + A_2(\sum_k \mathbb{V}_k - \mathbb{V}_t)^2 + A_3 \underline{\sigma}_{\mathbb{V}}^T \underline{\sigma}_{\mathbb{V}} + A_4 \sum_k \frac{1}{1 + e^{-\frac{(s_k - s_t)}{b}}}}$$

In the above expression, $\mathbb{V}_1$ is the first drop to reach the substrate, $\mathbb{V}_k$ is the volume of the k^{th} drop to reach the substrate, and $\mathbb{V}_t$ is the target drop volume we are attempting to achieve. s_k and s_t are measured and target minimum acceptable drop velocities, and $\underline{\sigma}_{\mathbb{V}}$ is the vector of volume measurement variances associated with each drop recorded. Each of these quantities represents a deviation from ideal performance, and the fitness therefore increases when such deviations are small. The coefficients A_i represent scaling factors, which balance and normalize the relationship between each type of deviation.

Each waveform is encoded into a binary string where binary representations of each waveform parameter or linear combinations thereof are concatenated together. Pairs of waveforms are stochastically selected for genetic recombination based on their fitness via fitness-proportionate selection, where the probability of selecting the j^{th} waveform is:

$$P_{\text{selection}}(j) = \frac{f_j}{\sum f_k}.$$

Selected waveforms are divided into pairs and used to create two new waveforms via two-point genetic crossover. The resulting waveforms are further modified by stochastic bitwise mutation. Additionally, a set number of top performing waveforms are automatically advanced to the next generation. By this process, a new queue of unique waveforms is generated for processing, and the optimization continues until a target number of waveform queue generations have been processed.

We selected the Microfab MJ-AT-01-050 piezoelectric inkjet, which consists of a circular

glass channel with a 50 μm aperture on one end and a threaded fluid fitting on the other end. The channel is actuated by applying a voltage to the radially aligned piezoelectric sheath wrapped around the tube. Figure 2 shows a cross-sectional diagram of the inkjet.

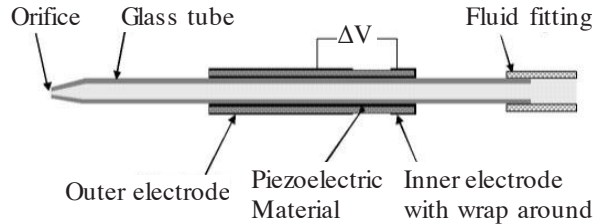


FIGURE 2. Cross-sectional diagram of Microfab MJ-AT-01-050 piezoelectric inkjet [7].

The JetDrive III inkjet controller generates unipolar and bipolar actuation waveforms. The unipolar waveform is represented by successive positive and negative voltage ramps of equal magnitude separated by a dwell time. We set the DC voltage for all waveforms to 0 V, so the unipolar waveform is represented by the four parameters, t_{rise} , t_{dwell} , t_{fall} , and V_1 . The duration of each ramp is defined by the rise and fall times. The bipolar waveform is represented by the seven parameters, t_{rise} , t_{dwell} , t_{fall} , t_{echo} , t_{fris} , V_1 and V_2 , as shown in Figure 3. The second voltage ramp is of greater magnitude than the first voltage ramp. A third ramp is introduced so that the voltage returns to DC after the echo time, has elapsed.

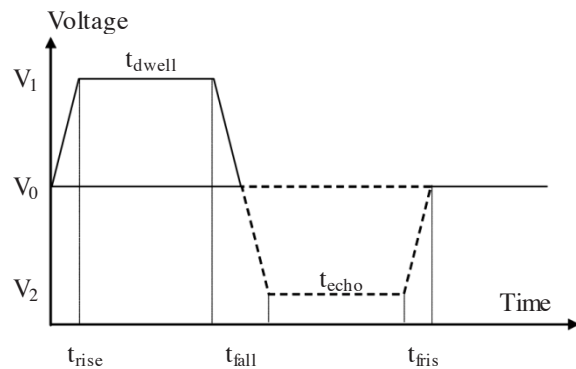


FIGURE 3. Diagram of unipolar (solid line) and bipolar (solid and dotted line) waveforms [8].

RESULTS AND DISCUSSION

A discussion of experimental results is provided next. Using a piezo inkjet with a 50 μm aperture, we have shown the ability to modify actuation waveforms to achieve specified target drop

volumes. While the search space is smaller for less complex waveforms, we have found that more complex waveforms are able to achieve drop volumes over a wider range as we attempt to generate smaller target volumes.

We tested the inkjet tuning system with ethyl acetate, which is considered to be a difficult material to inkjet due to its fluid properties. The Ohnesorge number and its inverse, the Z number, are considered important quantities that dictate the ability to inkjet a fluid with a particular inkjet[9, 10]. These numbers are described by the following relation, where ρ , σ , μ , and D represent the fluid density, fluid surface tension, fluid viscosity, and inkjet aperture diameter:

$$Z = \frac{1}{Oh} = \frac{\sqrt{\rho\sigma D}}{\mu}$$

Ethyl acetate is a fluid with ultra-low viscosity of 0.452 cP and low surface tension of 23.67 dyn/cm, which results in Z number of 51 for an inkjet with a 50 μm aperture. Many authors have suggested upper bounds on Z for inkjet printability, with the most recent and least conservative being approximately $Z < 43$ by Nallan for satellite related printability of low viscosity inks [11]. Therefore, automatically tuning an inkjet to jet this fluid at precise volumes should be considered challenging.

We set target volumes in increments of 10 pL and recorded the two best performing waveforms from each of two trials. We observe a trend where the smallest achievable target is extended by the use of more complex waveforms.

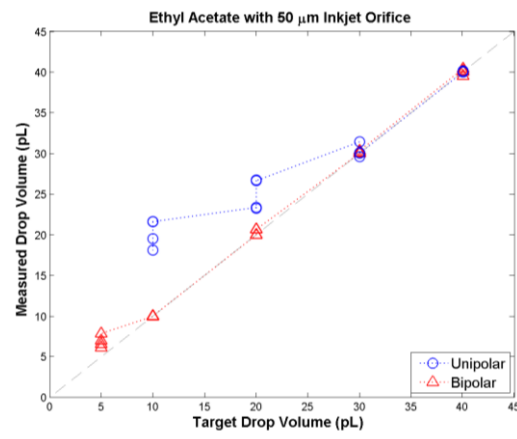


FIGURE 4. Inkjet tuning results for ethyl acetate with 50 μm aperture.

When tuning a 50 μm piezoelectric inkjet with ethyl acetate, the unipolar and bipolar waveforms were each able to accurately generate 30 pL and 40 pL drops. The unipolar waveform was unable to reach the target volume of 20 pL. Setting the target volume to 10 pL resulted in the generation of drops as small as 20 pL. This represents a tradeoff where increasing the gain in the volume error part of the fitness function adds selective pressure sufficient to produce drops closer to the boundary of achievable drop volumes. The bipolar waveform was able to generate 10 pL drops accurately, so we set the target volume to 5 pL and were able to generate 6.15 pL drops. To the best of our knowledge, we have not observed prior experimental evidence of such low drop volumes for a fluid outside the bounds of inkjetability, as defined by the Z number.

CONCLUSIONS

We have demonstrated a stochastic machine-in-the-loop inkjet control system for grayscale drop volume control. The system is able to test the performance of piezoelectric inkjet actuation waveforms and generate new waveforms in order to automatically generate drops of specified volumes. Higher order waveforms, which are more difficult to tune manually because of a larger number of tunable parameters, are shown to produce smaller and more precise drops than lower order waveforms, thereby increasing the inkjetability range of the system. These results are promising for applications in nanomanufacturing as automated inkjet tuning technology enables finer precision in process control.

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Uninterrupted Interpolation Of Short-Segmented Linear Tool-Paths Based On Jerk Limited Kinematic Corner Smoothing

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INTRODUCTION

This paper presents a novel “kinematics-based” approach to smoothen interpolation of short segmented tool-paths with controlled contouring errors.

CAD (Computer Aided Design) systems are used to design parts with high order smooth curves and splines. However, CNC (Computer Numerical Controlled) machines typically do not accept reference toolpaths defined in terms of splines and curves. As a result, CAM systems are used to discretize these smooth curves and generate tool-paths composed of linear point-to-point (P2P) position commands. As a matter of fact, continuous motion cannot be generated on a discrete tool-path. Currently, either geometric smoothing [1] or filtering techniques [2] are used to smoothen the reference tool-path, both of which have key limitations. For instance, geometric corner smoothing techniques are computationally stringent. On the other hand, filtering techniques cannot control the cornering error accurately.

Our previous research [3] overcomes some of these limitations by using a kinematic corner smoothing (KCS) approach. This method solves the cornering problem by smoothly blending the axis velocities between consecutive linear moves, and generate a time-optimal cornering motion by exploiting acceleration and jerk limits of drives while keeping the path errors constrained.

Nevertheless, previously proposed KCS technique [3] can only be applied to tool-paths that have long straight sections so that corners are not overlapped. This bottleneck limits the use of the KCS in practice. Practical machining tool-paths are composed of short linear segments, and thus the generated corners overlap each other.

In this paper, we extend the KCS approach to short segmented linear tool-paths by smoothly blending cornering velocity and acceleration “kinematically” from a position at the center of a corner to the center of the next corner based on the Jerk Limited Acceleration Profile (JLAP) [4].

KINEMATIC CORNER SMOOTHING (KCS)

The Kinematic Corner Smoothing (KCS) algorithm [3] solves the corner smoothing problem kinematically. It generates a symmetrical trajectory around the bisector of sharp corner by smoothly interpolating and blending kinematic boundary conditions of each axis, such as cornering velocity and acceleration. The generated trajectory becomes time-optimal because the KCS technique uses full potential of the drives’ acceleration and jerk limit.

This section introduces the basic idea of the KCS technique for single sharp corners, and extends it to short segmented linear tool-paths with a predefined cornering tolerance.

Basic idea of single corner smoothing

Figure 1 illustrates the strategy of the kinematic corner smoothing for the single sharp corner. As shown in Figure 1, the cornering acceleration (A_c) and velocity (V_c) at the entry and the exit sides of a cornering trajectory are identical. The acceleration vector is on the opposite direction of the velocity vector at the start of the corner, while the direction of both the acceleration and velocity vectors are on the same direction at the end of the corner. The relation between the path and axis accelerations can be expressed as:

$$\begin{cases} A_{s,x} = A_s \cos(\theta_1) = -A_c \cos(\theta_1) \\ A_{s,y} = A_s \sin(\theta_1) = -A_c \sin(\theta_1) \end{cases} \quad (1)$$
$$\begin{cases} A_{e,x} = A_e \cos(\theta_1 + \theta_2) = A_c \cos(\theta_1 + \theta_2) \\ A_{e,y} = A_e \sin(\theta_1 + \theta_2) = A_c \sin(\theta_1 + \theta_2) \end{cases}$$

where A_s and A_e are the acceleration boundary conditions at the start and the end of the cornering trajectory. Subscripts x and y describe Cartesian axes.

Similarly, the relational expression between the path and axis velocity can be expressed as:

$$\begin{cases} V_{s,x} = V_s \cos(\theta_1) = V_c \cos(\theta_1) \\ V_{s,y} = V_s \sin(\theta_1) = V_c \sin(\theta_1) \\ V_{e,x} = V_e \cos(\theta_1 + \theta_2) = V_c \cos(\theta_1 + \theta_2) \\ V_{e,y} = V_e \sin(\theta_1 + \theta_2) = V_c \sin(\theta_1 + \theta_2) \end{cases} \quad (2)$$

Here, V_s and V_e are the velocity boundary conditions at the start and end of the cornering trajectory.

Since boundary conditions at both sides of the corner is identical, generated trajectory becomes symmetrical at around the bisector of the corner. Therefore, the maximum cornering error occurs at the center of the cornering trajectory. The axis cornering acceleration and velocities are smoothly interpolated from the start to the end of the corner by the constant cornering jerk (J_C). Here, the cornering parameters (cornering acceleration, velocity, and time) are determined by blending the boundary conditions with constant jerk transitions expressed by following equations:

$$\begin{cases} A_{e,k} = A_{s,k} + J_{C,k} T_C \\ V_{e,k} = V_{s,k} + A_{s,k} T_C + \frac{1}{2} J_{C,k} T_C^2, \quad k = x, y \\ e_k = S_{mid,k} - P_{C,k} \end{cases} \quad (3)$$

where, A_s , A_e , V_s , and V_e are the boundary acceleration and velocity at the start and end of a cornering trajectory. ε is the cornering tolerance specified by the user. J_C is the jerk value of the interpolation, and T_C is total cornering time. Here, the cornering velocity V_C becomes the fastest cornering speed if the cornering jerk J_C is set as the maximum jerk value of the drive J_{max} . The cornering parameters can be calculated analytically as:

$$\begin{cases} J_C = J_{max} \\ A_C = 2^3 \sqrt{\frac{3\varepsilon J_{max}^2}{\sin\left(\frac{\theta_2}{2}\right) (\cos(\theta_1) + \cos(\theta_1 + \theta_2))^2}} \\ V_C = 2^3 \sqrt{\frac{9\varepsilon^2 J_{max}}{\sin^2\left(\frac{\theta_2}{2}\right) (\cos(\theta_1) + \cos(\theta_1 + \theta_2))}} \\ T_C = 2^3 \sqrt{\frac{3\varepsilon (\cos(\theta_1) + \cos(\theta_1 + \theta_2))}{J_{max} \sin\left(\frac{\theta_2}{2}\right)}} \end{cases} \quad (4)$$

As shown in Equation (4), the calculated value of the cornering acceleration (A_C) and velocity (V_C) ensure that the machine can change its feed direction smoothly within the user-given cornering tolerances (ε).

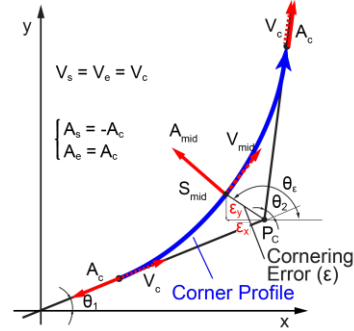


FIGURE 1. Kinematic Corner Smoothing (KCS) strategy.

Interpolation of Short Segmented Linear Tool-paths

Previous section presented corner smoothing for the local cornering case when corners are not within close vicinity of each other. In case of short segmented tool-paths, segment lengths are short. The blended corner trajectories overlap each other, which affects the cornering tolerance. This section applies kinematic corner smoothing when sharp corners are in close vicinity and overlapping. When the corners overlap the center of consecutive corner is connected based on the JLAP [4]. Kinematic conditions (acceleration, velocity, and position) at the midpoint(center) of a corner, is shown in Figure 1 as A_{mid} , V_{mid} , and S_{mid} respectively. They are calculated as:

$$\begin{cases} A_{mid,k} = \frac{A_{e,k} + A_{s,k}}{2} \\ V_{mid,k} = \frac{V_{e,k} + V_{s,k}}{4} \\ S_{mid,k} = P_{C,k} + \varepsilon_k \end{cases} \quad (5)$$

Figure 2 illustrates the strategy of continuously connecting consecutive corners on a short-segmented linear tool-path. As shown in Figure 2(b), acceleration values of the middle points of two corners are connected by piecewise constant jerk segments. In Figure 2, α_m depicts the transition point in the cornering time T_m . J_{m1} and J_{m2} are the jerk values of each jerk segment. The midpoint kinematics, i.e. displacement, velocity, and acceleration are solved through optimal selection of T_m , α_m , J_{m1} , and J_{m2} . The relational expression of midpoints of two corners can be written as the following:

$$\begin{cases}
A_{mid,k}^{ii} - A_{mid,k}^i = J_{m1} T_{m1} + J_{m2} T_{m2} \\
V_{mid,k}^{ii} - V_{mid,k}^i = A_{mid,k}^i T_{m1} + \frac{1}{2} J_{m1} T_{m1}^2 + (J_{m1} T_{m1}) T_{m2} \\
+ \frac{1}{2} J_{m2} T_{m2}^2 \\
S_{mid,k}^{ii} - S_{mid,k}^i = V_{mid,k}^i T_{m1} + \frac{1}{2} A_{mid,k}^i T_{m1}^2 + \frac{1}{6} J_{m1} T_{m1}^3 \\
+ \left(A_{mid,k}^i T_{m1} + \frac{1}{2} J_{m1} T_{m1}^2 \right) T_{m2} + \frac{1}{2} (J_{m1} T_{m1}) T_{m2}^2 \\
+ \frac{1}{6} J_{m2} T_{m2}^3
\end{cases} \quad (6)$$

where $T_{m1} = a_m T_m$ and $T_{m2} = (1 - a_m) T_m$. Equation 8 represents the position, velocity and acceleration kinematic compatibility conditions between consecutive corners. As noted, the 4 unknowns in Eq.8, namely T_m , J_{m1} , J_{m2} , and a_m must be solved from the 3 equalities. The free variable is used to find a solution which delivers jerk transitions (J_{m1} , J_{m2}) that within the limits of drives. The algorithm outlining planning of sequential cornering motion is presented below.

Step 1: Determination of cornering time:

The cornering time T_m between mid-points of corners is determined by selecting the axis that has the shortest cornering time. The other axis in cornering motion is called the "trailing axis".

Step 2: Calculation of the motion profile from the selected axis:

Motion profiles from the axis with the shortest cornering time T_m are calculated by solving the relational expression by setting a_m as arbitrary values.

Step 3: Calculation of the motion profile of trailing axis:

The motion profile of the trailing axis is calculated by solving the relational expression based on T_m set as the same value of the selected axis because the corner-profiling of both axes needs to finish at the same time.

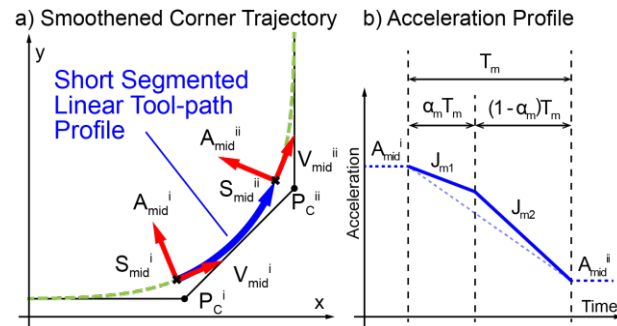


FIGURE 2. Strategy of KCS for short segmented linear tool-paths.

SIMULATION RESULTS

Figure 3 shows simulation results of the kinematic corner smoothing applied to short segmented linear tool-paths. As shown, the tool-path consist of 3 segments. The length of the first and third linear segment is 5 [mm] and the second linear segment is 0.3 [mm]. The cornering tolerance of both corners is set to 40 [μ m]. Since the cornering tolerance is large and the three linear segments are short, once corners are blended, their trajectories overlap. The programmed feedrate is 100 [mm/sec]. The acceleration and jerk limitations of each axis are set to $A_{max} = 4 \times 10^3$ [mm/sec²] and $J_{max} = 2 \times 10^5$ [mm/sec³].

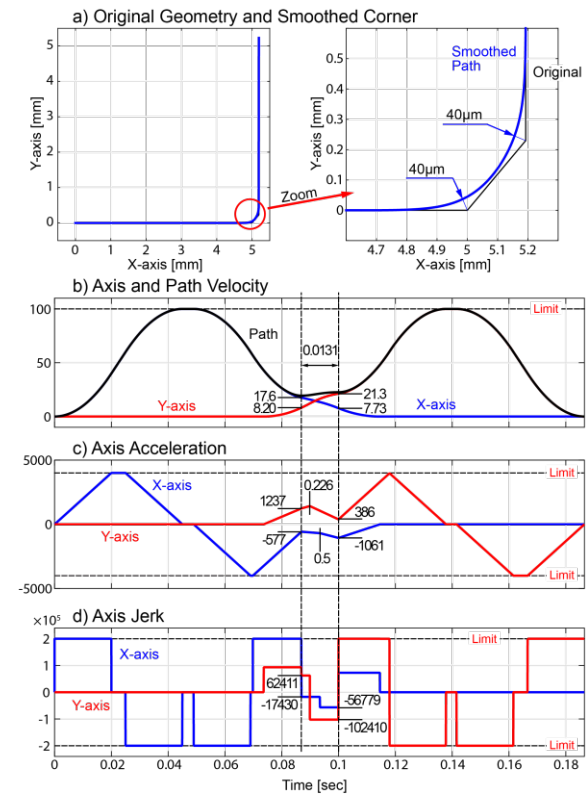


FIGURE 3. Interpolation of short segmented linear tool-paths using kinematic corner smoothing.

As shown in Figure 3, the generated trajectory is continuously interpolated by the short segmented linear tool-paths with specified contour error by connecting the midpoints of consecutive corners. On this instance, the cornering time of x-axis is selected for profiling the short-segmented linear tool-paths. So, the transition point of x-axis ($a_{m,x}$) is set as 0.5, and then the other parameters (T_m , $J_{m1,x}$, and $J_{m2,x}$) of the x-axis are determined. Finally, the

parameters of the y-axis ($\alpha_{m,y}$, $J_{m1,y}$, and $J_{m2,y}$) are calculated once the cornering time T_m is obtained.

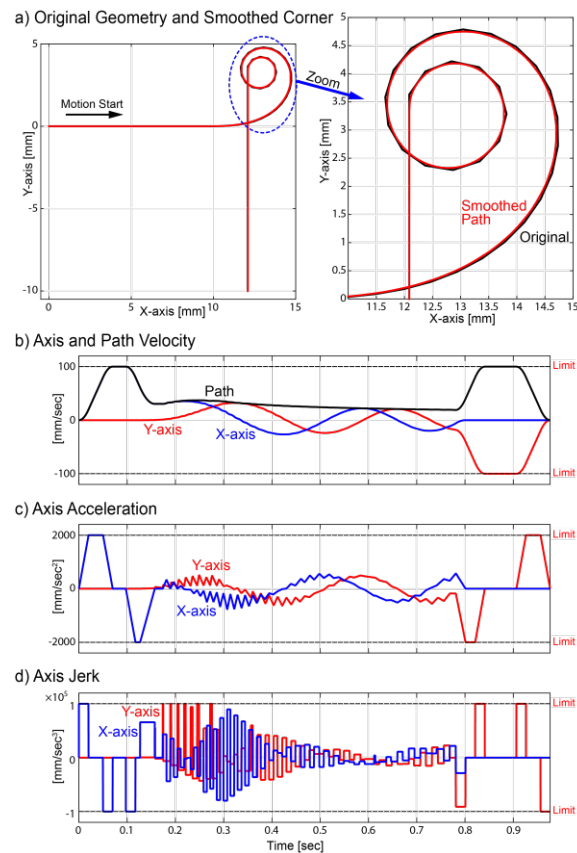


FIGURE 4. Interpolation of the spiral trajectory which is composed of the multi-segmented linear tool-paths using kinematic corner smoothing.

Next, the proposed KCS technique is applied on a multi-segmented linear tool-path shown in Figure 4. The programmed feedrate is 100 [mm/sec], and the cornering tolerance is set to 40 [μ m]. The acceleration and jerk limitation of each axis are set to $A_{max}=2 \times 10^3$ [mm/sec²] and $J_{max}=1 \times 10^5$ [mm/sec³].

Figure 4 depicts the proposed KCS algorithm successful smooth corners of the multi-segmented tool-path within the cornering tolerance (ϵ) and drives' limitations (A_{max} and J_{max}). Here, the cornering tolerances of the first 10 corners are decreased to reduce the cornering acceleration and velocity that ensures the interpolation within the drives' acceleration and jerk limitations.

One of the advantages of the proposed KCS technique is that the cornering velocity kept its speed during the overlapping corners. As shown

in Figure 4(b), the path velocity is maintained during the overlapping interpolation, and the velocity profiles of each axis are blended smoothly.

CONCLUSIONS

This paper proposed a new Kinematic Corner Smoothing (KCS) algorithm for the interpolation of short-segmented linear tool-paths, which generates cornering motion within user specified cornering tolerance and drive's limits. Overlapping corners composed of short-segmented linear tool-paths are smoothly interpolated to generate accurate and rapid cornering motion with the KCS technique. Simulation results show the potential of the proposed KCS technique to generate rapid trajectories along overlapped corners.

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ESTIMATION OF THE LOWER BOUND OF CONTROL BANDWIDTH REQUIRED IN THE CONCEPTUAL DESIGN OF THE WAFER STAGE

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INTRODUCTIONS

The chip manufacturing industry continually strives to pack more functionality into each integrated circuit (IC). This progress has had a major impact on the capacity of memory chips and the operating speeds of microprocessors [1]. The lithographic tool needs to be accurate to support the small feature sizes of the IC patterns that are being exposed. During exposure, the wafer and reticle are moved with constant velocity. The allowable position error during the scan is only a few nanometers [2]. Overlay is the ability of the lithographic tool to expose two images exactly on top of each other. A functional chip requires all chip layers to cooperate with each other. Overlay is directly influenced by stage-positioning accuracy, in addition to numerous other contributors. The average position error of the stage during exposure determines the location where the image is placed on the wafer. This measure is called moving average (*MA*). Moreover, the imaging process is degraded by a limited stage-positioning accuracy since position errors reduce contrast. Here, the moving standard deviation (*MSD*) of the positioning error during exposure is the critical measure [2].

The source of moving errors are disturbances from different resources in the system. Disturbance rejection is the major purpose for using negative feedback in most control systems. The effects of the disturbances are reduced when the feedback is negative (almost up to the control bandwidth) and increased when the feedback is positive (in the neighborhood of the control bandwidth). Proportional-integral-derivative (PID) controllers are standard tunable controllers in industry. The controller can be easily tuned to provide robust and fairly good performance for a great variety of plants [4]. However, the feedback integral over a frequency region where the feedback is negative and is equal to the negative of the integral over the range of positive feedback. Besides, the positive feedback is concentrated within a few octaves near the control bandwidth [5], the bandwidth

design is of extreme importance for disturbance rejection, and the design is also constrained by demand for the robust performance and system stability.

In the concept evaluation for the finestage design, it is vital to derive proper functional system-specifications. It is combined with assumed imperfections and disturbances that require to be translated into dynamics and control specifications. It is also essential to make some initial estimation about the required bandwidth of the controlled system, because this is a prerequisite for evaluating the influence of the dynamics of the mechanical system on servo stability [5]. Several methods are studied on the estimation of the required bandwidth. As in reference [6], Rankers proposed a simple method on the basis of the servo stiffness and the moving mass. This method neglected the effect of the derivative action of the controller and the disturbance was constrained to be the static force. As in reference [7], Ming estimated the bandwidth as $w_b = w_n / 5$, where the term

w_n stands for the first resonance frequency of the plant. This is not feasible for the concept evaluation of the fine stage, since the dynamic properties of the plant are to be determined based on the bandwidth requirement. Reference [8] provided a simple method to achieve high bandwidth while minimizing the closed-loop control cost. However, boosting the open-loop bandwidth will confront great obstacles in the wafer stage design due to the limitations on the mass and the heat dissipation of the moving stage. As in references [9,10,11], the control bandwidth was just designed to be as high as possible, which did not give the indication on how to derive the optimum control bandwidth satisfying various constraints.

In this paper, a new method of estimating the bandwidth in the concept evaluation of the wafer stage is proposed. It evaluates the effect of different resources of disturbances, and calculates the minimum bandwidth required to meet the demand for *MA* and *MSD*

specifications and the stability robustness. The purpose is to offer the basis for the design and evaluation of the dynamic properties of the system.

METHOD

MA and *MSD* are two critical specifications to evaluate the moving performance of the finestage in wafer stage. *MA* of the error is calculated as follows [2]:

$$MA(i) = \frac{1}{n} \sum_{j=i-n/2}^{i+n/2-1} e(j) \quad (1)$$

where the term e refers to the moving error. *MSD* of the error is calculated as follows [2]:

$$MSD(i) = \sqrt{\frac{1}{n} \sum_{j=i-n/2}^{i+n/2-1} (e(j) - MA(i))^2} \quad (2)$$

as shown in Figure 1[3], *MA* behaves like a low pass filter and it stands for the lower frequency properties of the error, *MSD* behaves like a high pass filter and it stands for the higher frequency properties of the error instead [3]. This indicates that the control bandwidth should be optimized to guarantee both specifications are satisfied. Meanwhile, since the plant gain tolerances increase with frequency, and the minimum necessary stability margins must be satisfied for the worst case, the minimum required control bandwidth should be constrained [12]. The aim of this paper is to develop a novel method to derive the lower bound of the control bandwidth required for both *MA* and *MSD* specifications and robust stability requirements in the concept evaluation of the wafer stage design.

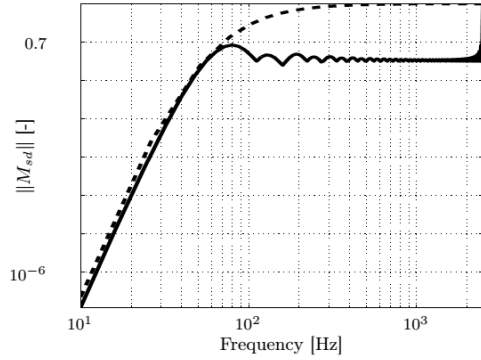
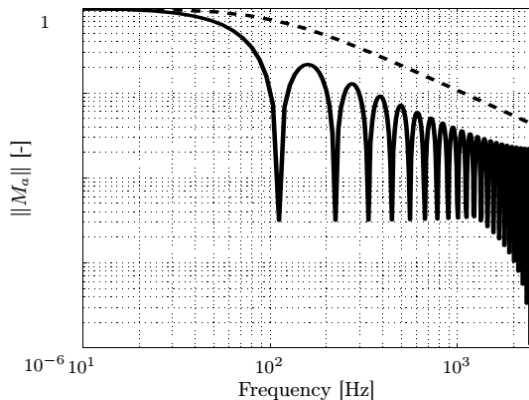


FIGURE 2. Spectrum of *MA* (upper) and *MSD* (lower) [3].

In the concept evaluation period, the simplified model of the closed loop of the wafer stage is sketched in Figure 2.

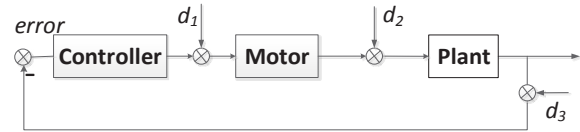


FIGURE 2. Simplified model of the closed loop of the wafer stage.

The parameters d_1 , d_2 and d_3 illustrate the disturbances acting on the system from different resources respectively. The disturbance models are derived though theoretical or experimental methods. For simplicity, the classical (proportional-derivative) PD controller is adopted for disturbance rejection, since the integral controller is able to eliminate steady-state errors, but it has little effect on transient response. Consequently, the integral controller is not applied here [12]. Stability design is prior in the closed loop design. Although gain and phase margins have been utilized to determine system stability, it is important to realize that the phase margin is the only reliable parameter on which to base such determinations [12]. The phase stability in the design of wafer stage is designed to be 60 degrees as a selection. As for a standard second-order closed-loop transfer function, the relationship between the phase margin PM and the damping ratio ζ is given by [12]

$$PM \approx \zeta \times 100 \quad (3)$$

When a zero or pole is present, there is no unique relationship between the phase margin

and the damping ratio. However, it is usually assumed that other system poles or zeros have little effect on the calculation of damping ratio [12]. The damping ratio is assumed to be 0.6 in the concept design of fine stage and the effect of this assumption will be examined later in the work. Since this paper aims to estimate the lower bound of the required control bandwidth in the concept design period, the plant could be simply treated as a moving mass. It is feasible since the fine stage exhibit a dominant rigid body behavior below the control bandwidth. However, this is not a necessary condition for this paper, the moving stage could be of more complex dynamic properties and the method will still work with more difficulties in optimization modeling and solving. The closed loop of the finestage can be presented as:

$$G_c = \frac{PAC}{1 + PAC} \quad (4)$$

in which

$$\begin{aligned} P &= 1 / ms^2 \\ A &= \text{constant} \\ C &= k_p + k_d s \end{aligned} \quad (5)$$

substitution of equation (5) into equation (4) gives

$$\begin{aligned} G_c(s) &= \frac{A(k_p + k_d s) / ms^2}{1 + A(k_p + k_d s) / ms^2} \\ &= \frac{A(k_p + k_d s)}{ms^2 + A(k_p + k_d s)} \\ &= \frac{A(k_p + k_d s)}{ms^2 + Ak_d s + Ak_p} \end{aligned} \quad (6)$$

and the damping ratio can be calculated as

$$\begin{aligned} \zeta &= \frac{k_d}{2} \sqrt{\frac{A}{mk_p}} = 0.6 \\ k_d &= 1.2 \sqrt{\frac{mk_p}{A}} \end{aligned} \quad (7)$$

The control bandwidth is commonly defined as the frequency range over a magnitude of the closed-loop frequency response. Note that this magnitude of the closed-loop frequency response does not drop more than 3 dB [13], e.g.

$$\begin{aligned} |G_c(jw_b)| &= \left| \frac{A(k_p + k_d * w_b j)}{-mw_b^2 + A(k_p + k_d * w_b j)} \right| = 0.707 \\ \frac{A\sqrt{k_p^2 + k_d^2 w_b^2}}{\sqrt{A^2 k_d^2 w_b^2 + (Ak_p - mw_b^2)^2}} &= 0.707 \\ \left(\frac{0.707m}{A}\right)^2 w_b^4 - 1.414 \frac{mk_p}{A} w_b^2 - 0.293k_p^2 &= 0 \end{aligned} \quad (8)$$

solution of equation (8) gives

$$\begin{aligned} w_b &= \sqrt{\frac{1.335Ak_p}{m}} \\ k_p &= \frac{mw_b^2}{1.335A}, k_d = \frac{1.0386mw_b}{A} \end{aligned} \quad (9)$$

In this way, terms k_p and k_d can both be expressed as the function of one single variable w_b which makes the optimization greatly simplified. The error caused by these disturbances is calculated as:

$$\text{error} = \frac{d_1 PA + d_2 P + d_3}{1 + PAC} \quad (10)$$

The optimization problem is to find the minimum required control bandwidth with constraints of *MA* and *MSD* specifications, i.e.

$$\begin{aligned} \text{Min } w_b \\ \text{s.t. } MA(w_b) \leq MA_0, MSD(w_b) \leq MSD_0 \end{aligned} \quad (11)$$

where MA_0 and MSD_0 stand for their specifications. The constraints of the optimization are strongly nonlinear. Nonlinear programming problems are often solved by methods of feasible directions [14]. These methods explore the feasible region by searching along directions which could reduce the objective function while maintaining feasibility. The design is changed along this direction until either a minimum of the objective function is found or a constraint is encountered [15]. The optimization involves the derivation of *MA* and *MSD* to the variable w_b . According to the definition of *MA* and *MSD*, the derivation can be derived as follows.

$$\begin{aligned}\frac{derror}{dw_b} &= \frac{-(2aw_b + bs)(d_3PA + d_2P^2A + d_1P^2A^2)}{(1 + PAC)^2} \\ \frac{dMA(i)}{dw_b} &= \frac{1}{n} \sum_{j=i-n/2}^{i+n/2-1} \frac{derror(j)}{dw_b} \\ \frac{dMSD(i)}{dw_b} &= \frac{1}{n} \sum_{j=i-n/2}^{i+n/2-1} \left(\frac{(error(j) - MA(i)) \left(\frac{derror(j)}{dw_b} - \frac{dMA(i)}{dw_b} \right)}{MSD(i)} \right)\end{aligned}\quad (12)$$

where the terms

$$a = \frac{m}{1.335A}, b = \frac{1.0386m}{A} \quad (13)$$

In the objective function and its derivation, the involvement of variable w_b means there exists an optimum value which could give the minimum objective function. Various methods can be adopted to solve this kind of optimization problem such as the method of Zoutendijk [16], the Frank-Wolfe algorithm [17], etc..

The method proposed above is used to estimate the lower bound of the required control bandwidth of a specified fine stage during the concept evaluation period. The schematic representation of the wafer stage is illustrated in Figure 3. The finestage floats above the coarse stage through a magnetic suspension system and is measured by the laser interferometer. The finestage is isolated from the metro-frame through active vibration isolation system. The balance mass is adopted to cancel the anti-force of the moving stage which could offer a quiet environment for the measuring system.

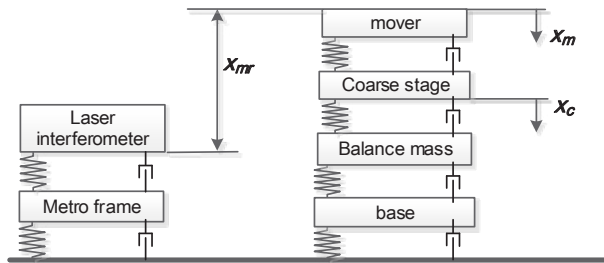


FIGURE 3. Schematic representation of the wafer stage.

The fine stage is treated as a moving mass of 10 kg in the concept evaluation period, the motor is

modeled as a static gain of 53.4579. The main disturbances affecting MA and MSD are the electrical noise from the driver, the D/A converter, the sensor noise and the disturbance force from the coarse stage. The closed loop system of the finestage with disturbances is sketched in Figure 4.

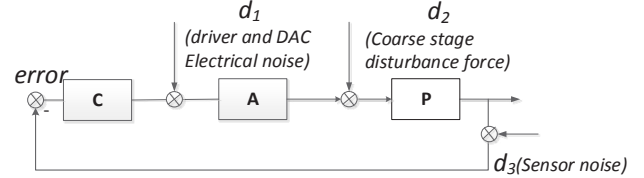


FIGURE 4. The closed loop system of the fine stage.

The electrical noise from the driver and the D/A converter was measured to have a standard deviation of 0.199 mV and had white noise characteristics. The PSD representing the driver and DA converter electrical noise is flat and ends up to the Nyquist frequency which is chosen to be 2500 Hz. The laser interferometer is chosen as a sensor, because it has a high resolution and a high precision. The sensor noise is typically 78 nm, concentrating in the frequency range below 10 Hz. The coarse stage affects the fine stage through the gravity suspension system, which could be modeled by a stiffness K and a damping coefficient C . The measured values are $K = 1520 \text{ N/m}$, $C = 242 \text{ N} \cdot \text{s/m}$, and the displacement of the coarse stage is supposed to be within 0.4 μm , which is derived through dynamic error budget. With the disturbance models derived, the lower bond of the required control bandwidth is designed using the optimization model proposed above to satisfy the specifications below.

$$\begin{aligned}MA &\leq 2\text{nm} \\ MSD &\leq 5\text{nm} \\ \text{phasemargin} &= 60 \text{ degrees}\end{aligned}\quad (14)$$

The lower bound of the optimized control bandwidth is calculated to be 252 Hz, MA and MSD with the optimized control bandwidth are illustrated in Figure 5. It shows that both of them satisfy the requirement for performance specifications. The phase margin of the open loop is 59.881 degrees, which also meets our requirement for system stability. However, it is a little smaller than the designed value of 60

degrees because of the existence of the extra zero introduced by the controller.

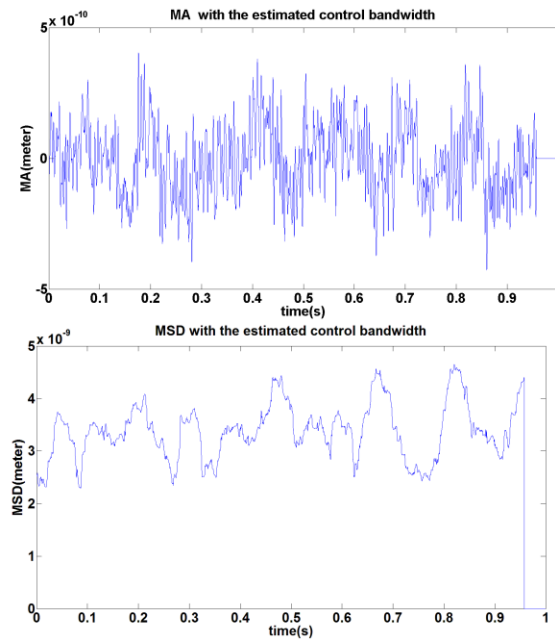


FIGURE 5. MA and MSD with the estimated control bandwidth.

CONCLUSION

The lower bound of the required bandwidth of the controlled system is needed in the concept evaluation to offer the basis for the design and evaluation of the dynamic properties of the system. The performance of the wafer stage should satisfy *MA* and *MSD* specifications which are inherently contradictory and should have enough stability robustness. Therefore, the control bandwidth should be optimized to guarantee both the performance and the robust stability. This paper proposed a new method of deriving the lower bound of the required control bandwidth in the concept evaluation of the wafer stage considering different resources of disturbance. It established the optimization model related to the single variable of the control bandwidth which simplified the optimization problem substantially. The effectiveness of this method was demonstrated by the application to the design of a specified fine stage. Results showed that the optimized control bandwidth guaranteed the *MA* and *MSD* specifications as well as the phase margin. This method has its potential to be applied to other moving stages with more complex dynamic properties which will be the further objective of our research.

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DESIGN OF POSITIONING CONTROL SYSTEM OF OPPOSED PAD WATER HYDROSTATIC THRUST BEARINGS

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INTRODUCTION

In order to improve machining accuracy of the precision machining such as the single point diamond cutting, precise rotational motion of the machine tool spindle is particularly important.

As well known, the constant pressure type of the hydrostatic bearings is commonly used for many applications, including machine tool spindle, because the hydrostatic bearings achieve precise motion.

In this case, however, increased external forces acting on the bearings are supported by adjusting bearing forces generated by changing bearing gap, namely bearing displacement. Based on the operating principle of the constant pressure type of the hydrostatic bearings, thus, the resultant displacement of the bearing due to the change of the external forces cannot be avoided.

To cope with the problem, this study deals with displacement control of an opposed pad type of the water hydrostatic thrust bearings^[1]. The displacement control system of the water hydrostatic thrust bearing uses an electrically controllable flow control valve.

If the displacement of the hydrostatic bearing can be controlled according to the command signal into the designed controller, it achieves not only to maintain the original position of the bearing but also to control arbitrarily the displacement in response to the required displacement. If the controller maintains the original bearing position regardless of the change of the external forces, infinity bearing stiffness can be made. In contrast, if the bearing displacement can be controlled according to the control signal, the bearing can be used for an actuator with excellent resolutions^[2]. For instance, the small depth of cut for the diamond turning will be controlled by the control system. Furthermore, the thermal error of the spindle or

machine tool will be compensated by the control system, as well.

In the following parts of the paper, design of a feedback control system of the water hydrostatic bearings using a flow control valve will be presented. Then the control performances will be compared depending on different valve layouts; the series connection and the parallel connection with the bearing restrictor. Positioning accuracy of the designed control system and control performances against the external forces will also be investigated experimentally.

NOMENCLATURE

$\hat{u}$	: Applied voltage to flow control valve
$\hat{h}$	: Bearing displacement
$\hat{q}_{c1}, \hat{q}_{c2}$	: Flow rate in bearing restrictor
$\hat{q}_{b1}, \hat{q}_{b2}$	: Flow rate in bearing surface
$\hat{p}_{r1}, \hat{p}_{r2}$	: Recess pressure
$\hat{F}_c$	: External load
k_{c11}, k_{c12} k_{c2}	: Inverse of flow resistance of bearing restrictor
k_1	: Gain of flow control valve between input voltage and flowrate
k_2	: Pressure - flow rate gain of flow control valve
k_b	: Inverse of flow resistance of the bearing
k_q	: Gain of bearing surface to determine flowrate
m	: Number of recesses
A	: Area of bearing pad
$\bar{A}$	: Coefficient of effective area of bearing pad
M	: Mass of spindle rotor
c	: Viscous resistance
k	: Bearing stiffness

- R_{s11}, R_{s12} : Gain of the recess pressure, applied voltage and bearing displacement (series connection)
 R_{s2}
 R_{p11}, R_{p12} : Gain of the recess pressure and applied voltage, bearing displacement (parallel connection)
 R_{p2}
 K_p : Proportion gain
 K_i : Integral gain

STRUCTURE AND BASIC PERFORMANCE OF A WATER HYDROSTATIC SPINDLE

The opposed pat type of the water hydrostatic thrust bearing that is used in this study is fabricated in the spindle as presented in Fig. 1. A flow control valve that is used to control bearing displacement is placed outside of the spindle. The flow control valve controls the flow rate of the left-hand side of the bearing surfaces. Although the flow control valve is used for the one side of the bearing surface in this case, however it is possible to control flow rate for both bearing surfaces by putting another control valve to the right-hand side of the bearing surface, as presented in Fig. 1.

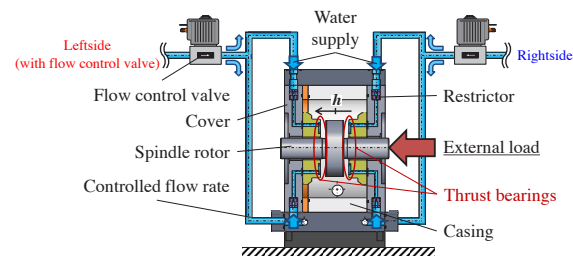


FIGURE 1. Structure of water hydrostatic spindle with flow control valve

OPERATING PRINCIPLE OF BEARING DISPLACEMENT CONTROL

The constant-pressure type of the hydrostatic bearings is generally used in many precision machine applications. In this case, the recess pressure in order to sustain the external load acting on the bearings is adjusted by changing bearing gap. As a result, the bearing displacement is changed according to the magnitude of the external forces.

Meanwhile, the recess pressure can also be controlled by changing the flowrate of the lubricating fluid into the bearing surfaces so that the desired bearing displacement can be obtained. Various flow control methods have been studied by many researchers. In this study, we use commercialized electrically controllable flow control valve. The range of the

displacement control is within the gap size of the bearing, namely from several micrometers to several tens of micrometers. This increases positioning resolution of the control system.

The displacement control system gives two benefits. First, the control system makes the bearing stiffness infinite by setting desired zero displacement in the control system. Second, the control system can be used as an actuator to change for instance depth of cut for machining applications. Furthermore, the displacement control system can be used for compensating thermal deformation of the machine tool, as well. Layout of flow control valve used in this study and bearing restrictor is presented in Fig. 2. As shown in Fig. 2 (a), when a flow control valve is used in the series connection, the flow control valve is connected in series to the constant restrictors. In contrast, as shown in Fig. 2 (b), in the parallel connection, the flow control valve is connected in parallel to the constant restrictors. Both connections of the flow control valve were investigated experimentally.

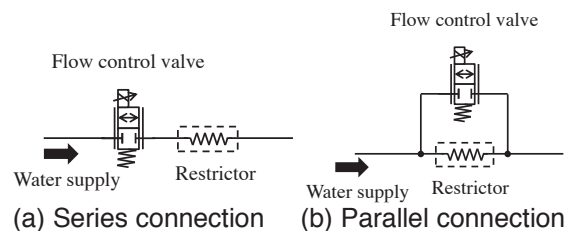


FIGURE 2. Layout of flow control valve

Basic structure of the flow control valve used in this study is presented in Fig. 3. The plunger of the valve moves upward and downward according to the applied control voltage, changing opening flow area of the valve.

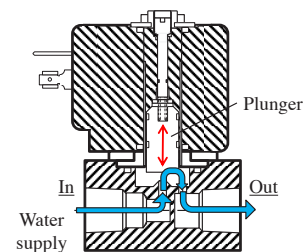


FIGURE 3. Structure of flow control valve

In the series connection, the flow control valve and conventional bearing restrictors are connected sequentially. Thus, constant initial voltage must be applied to the flow control valve

connection and parallel connection are presented in Eq. (1) and Eq. (2), respectively.

$$G_{series}(s) = \frac{-m\bar{A}R_{s11}}{Ms^2 + cs + k - m\bar{A}(R_{s12} + R_{s2})} \quad (1)$$

$$G_{parallel}(s) = \frac{-m\bar{A}AR_{p11}}{Ms^2 + cs + k - m\bar{A}A(R_{p12} + R_{p2})} \quad (2)$$

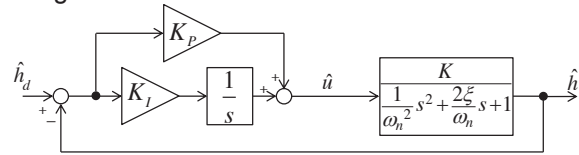
Design of feedback control system

In this study, a PI feedback control system is designed for the bearing displacement control system. The block diagram of the designed feedback control system is presented in Fig. 6. In this control system, the bearing displacement is measured using the laser displacement sensor. Control voltage into the flow control valve was applied via D/A converter through valve driver. In order to determine the model parameters and the control gains, step response of the open loop control system was measured by applying step input voltage into the flow control valve. Based on the measured step response, the transfer function of the system was determined. Then, the transfer function was used to determine control gains of the feedback control system. As a result, when the flow control valve is used in the series connection, proportional gain K_{Ps} and integral gain K_{Is} were determined to be 0.2 V/ μm and 5.1 Vs/ μm , respectively. In addition, when the flow control valve is used in the parallel connection, proportional gain K_{Pd} and

[illegible]

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integral



ga in K_p were 0.6 V/ μm and 12 Vs/ μm , respectively. After this, control performance of the series connection and the parallel connection are investigated experimentally.

FIGURE 6. Block diagram of feedback control system

STATIC CHARACTERISTIC OF DEVELOPED FEEDBACK CONTROL SYSTEM

Static characteristic of the designed feedback control system was measured when the flow control valve are used in the series connection and the parallel connection, respectively. The schematic drawing of structure of the control system when the flow control valve is connected in the series connection is presented in Fig. 7. In the parallel connection, the fundamental structure of the control system is same as Fig. 7, except for the valve layout.

In the experiments, the flow control valve was used in the both sides of the bearing surfaces as Fig. in Fig. 7 for future experiments. In the series connection, constant voltage was applied into the flow control valve in the right-hand side of the bearings. Then constant desired bearing displacement signals were input from 0 nm to 250 nm with step size of 50 nm into the flow control valve of the left-hand side of the valves. In the experiment, no external load was applied to the bearing.

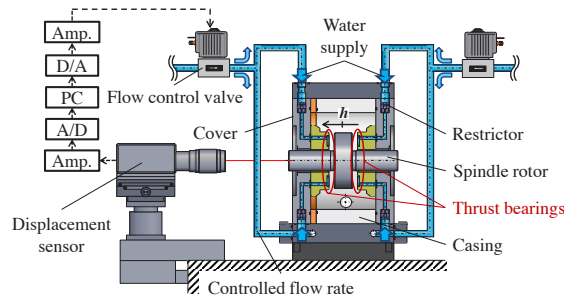
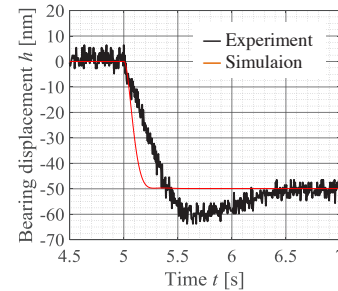


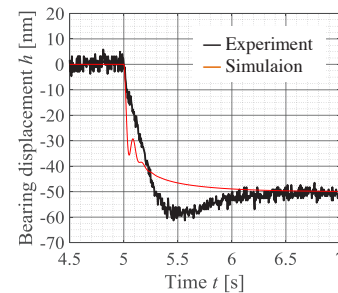
FIGURE 7. Experimental device of static characteristics

Experimental results are presented in Fig. 8. In Fig. 8 (a), the step response of the control

system with the series connection is presented. In contrast, Fig. 8 (b) shows the step response of the control system with the parallel connection. Comparing both results, significant difference in the overshoot and the settling time are not confirmed. The overshoot was approximately 20 %, the settling time was less than 1.5 seconds.



(a) Series connection



(b) Parallel connection

FIGURE 8. Experimental results of step response

In addition, the static characteristics of the designed control system were measured and then compared depending on the valve layout. The results are shown in Fig. 9.

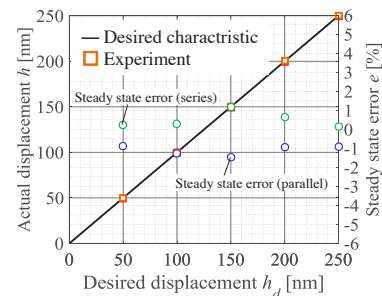


FIGURE 9. Static characteristic (Stepwise by 50 nm)

In the experiments, the desired constant displacement signals from 0 nm to 250 nm with

50 nm step were input to the control system. As shown in Fig. 9, it is indicated that the steady state error was less than 1.5 %.

From the experimental results, it is verified that the control system can control the bearing displacement with 50 nm resolutions.

Next, desired constant displacement signals were input from 0 nm to 50 nm with step size of 10 nm. In the experiment, no external load was applied to the bearing as well. Experimental results are presented in Fig. 10. From these results, it is verified that the designed control system can be control with nanometer resolutions.

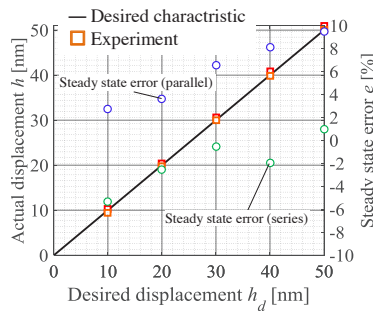


FIGURE 10. Static characteristic (Stepwise by 10 nm)

The experimental results indicate possibility that the designed control system can use as the actuator to control small depth of cut for machining applications. In addition, the designed control system can be a compensation tool for thermal deformation of the spindle and machine tool.

INFLUENCE OF EXTERNAL LOAD ON BEARING DISPLACEMENT

Dynamic characteristics of the designed control system were investigated when the external step load was applied to the spindle in the axial direction. In the experiment, the flow control valve was used in the series connection and the parallel connection, respectively. The schematic drawing of the control system when the flow control valve is connected in the series connection is presented in Fig. 11. In the parallel connection, the fundamental structure of the control system is same as Fig. 11, except for the valve layout.

In the experiments, the external step load of approximately 1 N was applied to the bearing in the axial direction, because cutting resistance is in general at most approximately 1 N during

single point diamond cutting. In the experiments, the desired bearing displacement signal was set to be 0 nm. The bearing displacement was then measured by the laser displacement sensor as Fig. 11.

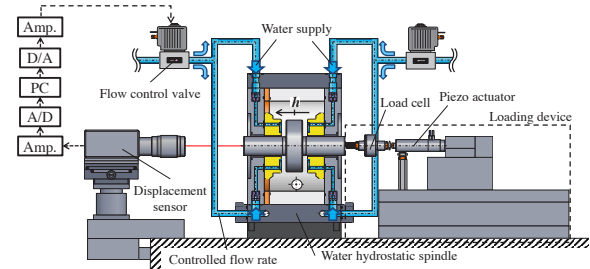
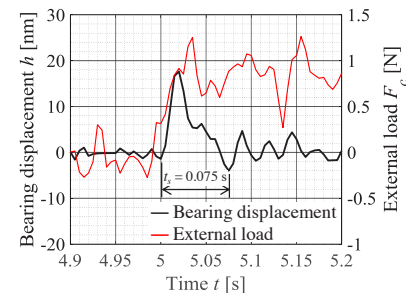
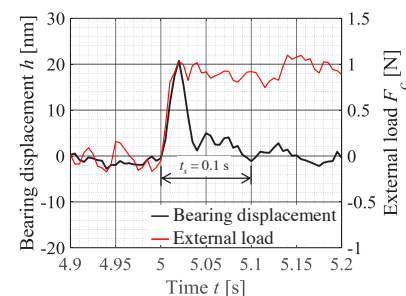


FIGURE 11. Experimental device of measuring response against external loads

Experimental results are presented in Fig. 12. In Fig. 12 (a), the response of the control system with the series connection is presented. In contrast, Fig. 12 (b) shows the characteristics of the control system with the parallel connection. In both results, significant difference in the bearing displacement and the settling time are not confirmed. As a result, the maximum bearing displacement was approximately 20 nm, the settling time was less than 0.1 second.



(a) Series connection



(b) Parallel connection

FIGURE 12. Responses of designed control system against stepwise external load.

From these experimental results, it is verified that the designed control system can compensate the resultant bearing displacement due to the external loads such as the cutting forces.

In addition, the compensation range against external loads was evaluated experimentally when the flow control valve is used in the series connection and the parallel connection, respectively. In the experiments, the external load was applied from 0 N to 1000 N to the rotor in the axial direction. The desired bearing displacement was set to be 0 nm. Experimental results are presented in Fig. 13. As shown in Fig. 13, it is verified that the control system can fully compensate the influence of the external loads. Accordingly, the zero bearing displacement was maintained in the range to the external loads of 1000 N.

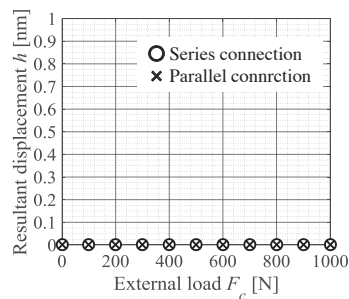


FIGURE 13. Controllable range of external loads

CONCLUSIONS

In order to improve the machining accuracy of the precision machining, the control of the constant pressure type of the water hydrostatic thrust bearing was investigated in this study. In principle, the resultant displacement of the constant type of the hydrostatic bearings due to the change of the external forces is inevitably avoided. To cope with the problem, a bearing displacement control system was designed and tested. In order to control the flowrate into the bearings, an electrically controllable flow control valve was used. Feedback control system was then designed with different valve layouts; the series connection and parallel connection.

Static characteristics and dynamic characteristics of the designed feedback control system were investigated experimentally for both valve layouts.

In the experimental results on the static characteristics, significant differences in the performances such as the overshoot and the settling time were not observed in both valve

layouts. In addition, the steady state error was less than 0.2 nm.

In the tests of the dynamic characteristics, the settling time was less than 0.1 second when the external step load of approximately 1 N was applied. The results indicate that the designed control system can compensate the resultant bearing displacement due to external loads.

In the present study, the flow control valve connected with one side of the bearing surfaces was controlled. In future work, in order to improve the control performance, two flow control valves will be controlled simultaneously with the feedback control system to be designed.

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VIBRATION ATTENUATION BY EDDY CURRENT WITH DIFFERENT COMBINATIONS OF MAGNETS.

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INTRODUCTION

When a conductor is vibrating in the presence of a magnetic field, in a way that the magnetic lines are perpendicular to the surface area of the conductor and are continuously changing with time, an eddy current is induced in the conductor [1]. This current always opposes the change in magnetic flux thus causing a damping effect to bring the vibrating body to a halt.

If the magnetic flux is equal to magnetic field (B) times the area (A):

$$\Phi = B \cdot A$$

Then according to Faraday's – Lenz's law induced emf will be:

$$\mathcal{E} = - \frac{\Delta \Phi}{\Delta t}$$

The negative sign in the equation indicates that the magnetic field induced by the current opposes the magnetic flux producing it.

Eddy current have significant effect in attenuation of vibration [2]. The objective of this work is to develop a model to simulate the effects of eddy currents on vibrations. The designed model can be used to damp the vibrations in a structure without the use of a damper in physical contact with it [3]. The eddy current is induced into the vibrating body using neodymium magnets. The Damping effects of this method are analyzed by varying the magnetic field under different combinations of magnets [4] and the distance between the magnet and the vibrating body.

SIMULATION

For the simulation COMSOL MULTIPHYSICS software is used. A (330mm x 30mm x 6mm) Aluminum beam is supported from one end and on the other end 10N impact force is applied. The force cause the beam to vibrate. Meanwhile a Neodymium magnet (Nd₂Fe₁₄B) is placed at the distance of 2mm from the suspended end of the

beam. This magnet has Coercivity of about 7MA/m (as one neodymium magnet have coercivity about 1.5MA/m and we are using two magnets on the axial side of beam and two on the parallel side of the beam, so the estimated collective coercivity is 7MA/m). Now as the beam vibrates the number flux lines of the magnet through the beam changes that cause eddy current to generate in the beam. This eddy current cause the damping effect in the beam due to which the beam stops vibrating eventually [5]. *FIGURE 1* shows the displacement frequency response of the beam when it is vibrating with natural frequency and when it vibrates under the damping effect. It is clear that the damping caused by eddy current results in about 80% attenuation of vibration in Al-beam.

As the results by the simulation are good enough to test them in real, so a customized setup is devised to verify the damping effect.

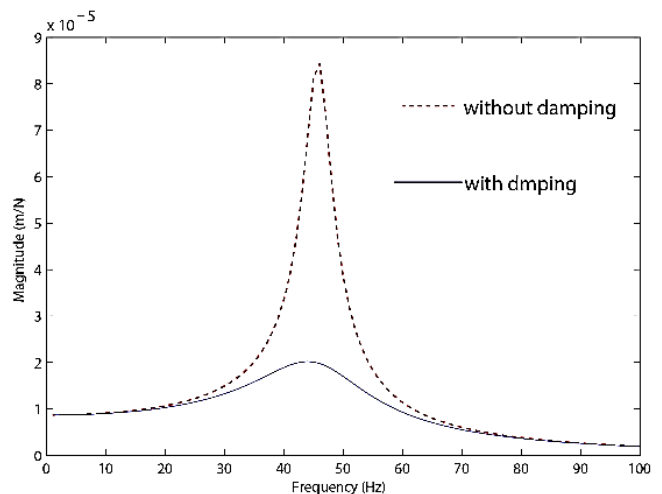


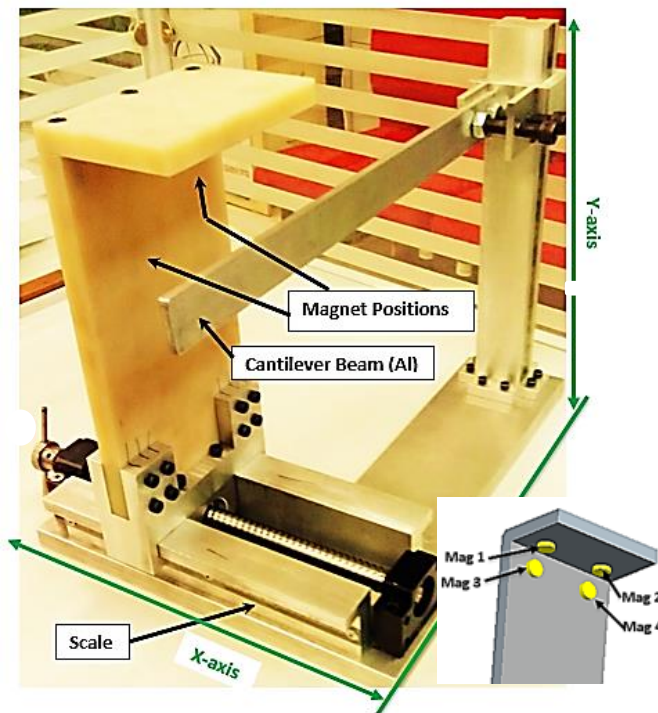
FIGURE 1: Displacement frequency response (m/N) of beam under no damping and when magnet is placed at 2mm distance.

EXPERIMENTAL SETUP

In this experiment, the neodymium magnets are mounted on a teflon structure and the aluminum beam (330mm x 30mm x 6mm) is mounted at on

an X-Y plotter mechanism to adjust its distances from the magnet. For horizontal (X-axis) adjustment a ball nut with lead screw is attached to the magnet mount and for vertical (Y-axis) adjustment aluminum beam is attached with a graduated scale *FIGURE 2*. Two scales with the sensitivity of 0.5mm are attached to X and Y axes for precise adjustment.

FIGURE 2. Experimental hardware



There are four distinct positions for the magnets. A pair of magnets is placed in front of the beam, such that its magnetic field lines pass through the beam perpendicular to the face with greater surface area. While another pair is placed above the beam, such that magnetic field lines pass through the beam perpendicular to the face with a smaller surface area *FIGURE 2*.

The distance between the beam and the magnets is adjusted by the ball screw and results are collected at 0.5mm, 1.0mm, 1.5mm and 2mm.

EXPERIMENTS

The numbering of the magnets is in such a way that the magnet mounted above the beam on our left side (while facing the apparatus) is magnet number 1 and the one on right side is magnet number 2. Similarly for the pair of magnets in front of the beam, the one on the left side is magnet number 3 and the one on the right side is magnet number 4. Then the seven combinations on which the experiment is performed are in TABLE 1.

TABLE 1. The seven different combinations of four magnets are shown. Where "Mag" is for magnet, "S" is for South Pole of the magnet, "N" is for North Pole of the magnet and "X" represents when there is no magnet at that place.

Combination	Mag1	Mag2	Mag3	Mag4
1	X	X	X	X
2	S	S	X	X
3	S	S	S	S
4	X	X	N	S
5	X	X	S	S
6	N	N	S	S
7	N	S	S	N

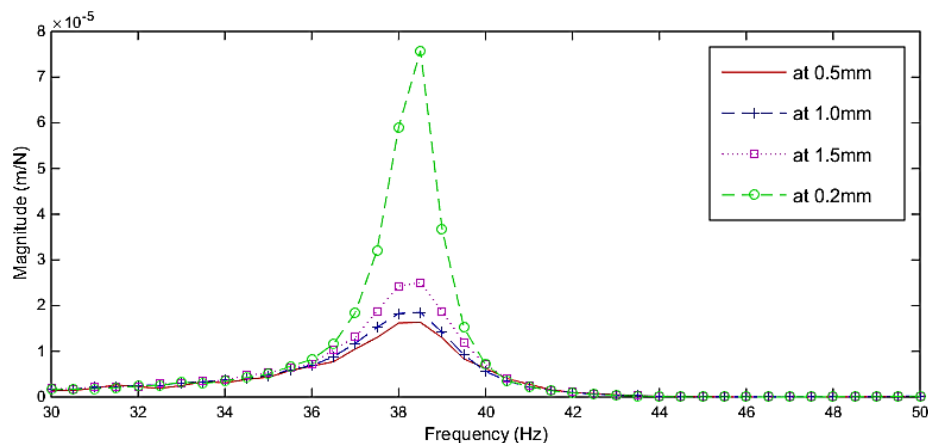


FIGURE 3. Displacement frequency response (m/N) of beam with combination #7 of magnets at 0.5mm, 1.0mm, 1.5mm, 2.0mm.

Now the experiments are performed by a hammer impact test in which we use the hammer (PCB-086C03) to strike the Al-beam to generate vibrations and an accelerometer (KISTLER-8778A500) with sensitivity of 11.11 mV/g, is placed near the impact point of the hammer to calculate the displacement of the beam. Both accelerometer and impact hammer are attached to a data acquisition card that feeds the data to the CutPro V9.3 software which compares the impact force of the hammer and the displacement from accelerometer and generates a FRF graph *FIGURE 3* and calculates the damping ratio of the beam.

CALCULATIONS AND RESULTS

Seven combinations of magnets are then tested at four different distances from the beam in *FIGURE 4* at 0.5mm, *FIGURE 5* at 1.0mm, *FIGURE 6* at 1.5mm and *FIGURE 7* at 2mm. As the first mode of vibration occurred at 38.36Hz so the following graphs shows the displacement of the beam at this frequency.

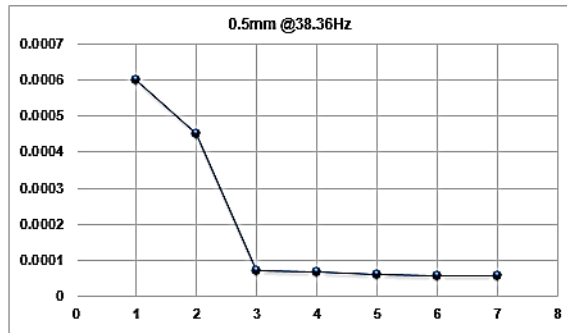


FIGURE 4. Displacement frequency response (m/N) of beam on vibration at seven different combinations of magnets. The experiment is performed at 0.5mm.

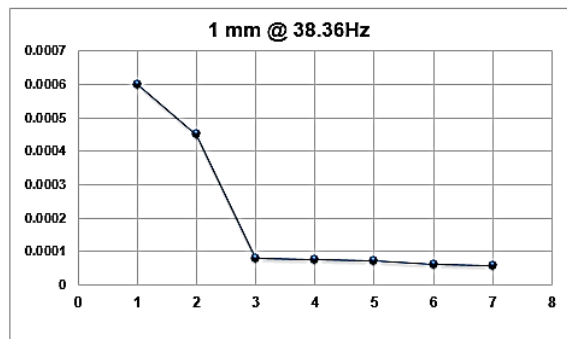


FIGURE 5. Displacement frequency response (m/N) of beam on vibration at seven different combinations of magnets. The experiment is performed at 1.0mm.

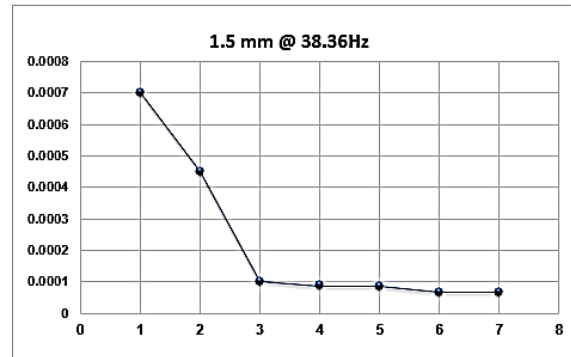


FIGURE 6. Displacement frequency response (m/N) of beam on vibration at seven different combinations of magnets. The experiment is performed at 1.5mm.

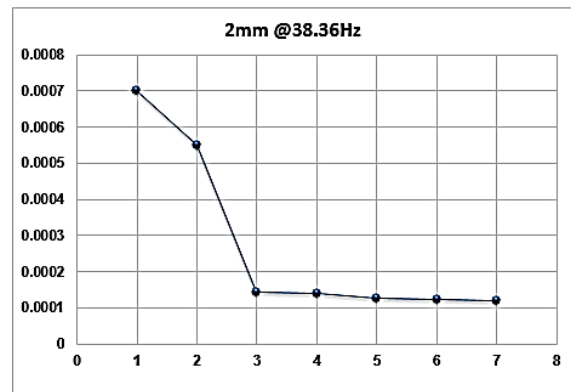


FIGURE 7. Displacement frequency response (m/N) of beam on vibration at seven different combinations of magnets. The experiment is performed at 2.0mm.

From the above four displacement graphs it is clear that the damping force decreases with the increase in distance between the beam and the magnets. Moreover the combination and pole side of the magnets also have an effect on attenuation of vibrations and the best combination proved to be the seventh one which showed the most effective damping.

As the seventh combination is more effective than the others so the following graph shows the damping ratios of combination seven setup at four different distances from the beam.

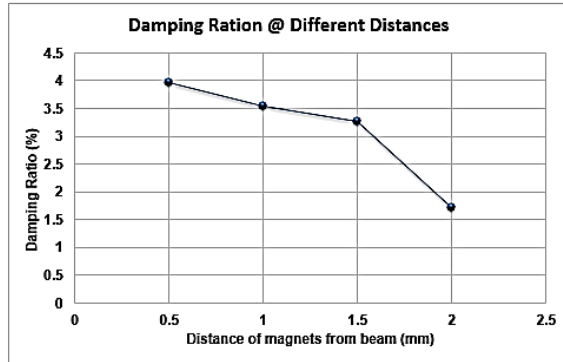


FIGURE 8. The damping ratios (%) of beam with combination # 7 of magnets at 0.5mm, 1.0mm, 1.5mm and 2mm.

From the graph it is clear that the damping ratio decreases with the increase in distance between beam and the magnets.

COMPARISON AND CONCLUSIONS

On comparing the both simulation and real test results we came to know that except from a slight change of frequency, all the other results are similar with one another.

On observing that different combinations of magnets have different effects on the damping ratios. The optimum combination may reduce the number of magnets required to achieve a specific amount of damping. On the other hand the distance between the magnets and the beam shows an inverse relation with the damping ratio. The smaller the distance more effective the damping will be.

By analyzing the experimental results, we can conclude that vibration attenuation with eddy current is a very effective method not just because it is a non-contact damper but also because it, naturally, controls the amount of force and direction of force required to reduce the vibration with great accuracy and speed.

In future, this work will be extended by using greater variations in distance and introducing more combinations. Another line of research would be to validate the linearity of the force due to the eddy current, acting against the vibration velocity with respect to the magnetic intensity and the distance between the beam and the magnets. Moreover electromagnetic shaker will be used instead of impact hammer for more real and continuous vibrations as it have controlled input signal that can generate desired amount of impact. It is also to be verified that the amount of induced eddy current in a beam directly depends on the intensity of vibration in the beam and the

size of the magnetic flux [6]. Finally we will also perform finite element simulation of the above mentioned experiments.

ACKNOWLEDGEMENT

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MITIGATION OF RESIDUAL VIBRATION IN PRECISION SCANNING STAGES USING ASSIST DEVICES

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OVERVIEW

Passive pneumatic vibration isolators are widely used in precision scanning stages to suppress ground vibration, but they contribute to long settling times of position errors due to the residual vibration created by inertial reaction force disturbance of linear motors. Residual vibration must be mitigated to ensure better tracking, especially when high accelerations and decelerations of stages are employed to boost throughput. This paper presents a residual vibration mitigation method, which utilizes optimally controlled assist devices, considering the dynamics of the isolated base. The effectiveness of the proposed method is shown using simulations based on parameters of a magnet assisted scanning stage prototype.

INTRODUCTION AND MOTIVATION

Precision scanning stages are indispensable in advanced manufacturing processes like silicon wafer patterning, inspection and packaging [1 – 4]. They are designed to provide highly precise motions; therefore, ground vibration is prevented from reaching the vibration-sensitive sensors and tools using pneumatic isolators (i.e. soft springs) installed on machine base [5]. In order to boost throughput, high accelerations and decelerations are employed at each end of scan stroke to minimize non-production time. However, the presence of large inertial motor reaction force produces residual vibrations of the isolated base. With negligible frictions present in guiding mechanisms commonly employed in precision scanning stages (e.g. magnetic or air bearings [1, 2, 4]), tracking errors proportionally increase with the residual vibration of the machine base. To ensure the positioning precision, the residual vibration must be mitigated.

Prior work by the authors showed that permanent magnet (PM) based assist devices, mounted on the machine ground of precision scanning stage, can reduce the motor force required to drive the table [1]. In this prior work,

the PMs were optimally controlled to minimize the actuation effort (i.e. the 2-norm of the linear motor force), as an indirect way to reduce residual vibration. In this study, the PMs are controlled to directly reduce residual vibration predicted using a model of the isolation system dynamics. Using simulations carried out on a scanning stage prototype [1]. The proposed approach is shown to be more effective in reducing residual vibration compared to the authors' prior approach of minimizing motor forces.

MAGNET ASSISTED STAGE

The schematic of magnet assisted stage is shown in Figure 1 to provide some context for the paper. The focus of this work is only on the scanning (i.e. x -axis) direction as shown in the figure. For more details on the stage design, readers are referred to the prior work by the authors [1].

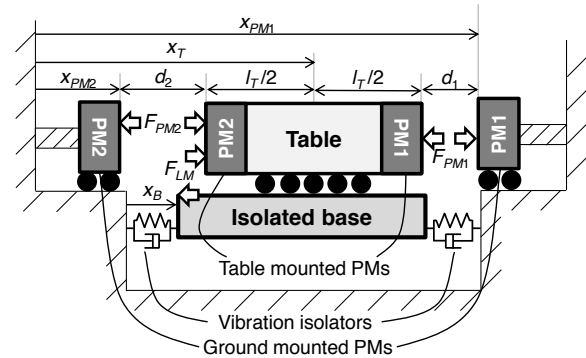


FIGURE 1. Scanning stage with PM based assist devices providing assistive force from machine ground.

The vibration isolators denoted with springs and dampers provide ground vibration isolation, while the linear motor force (F_{LM}) excites the isolated base, creating the base displacement (x_B). Each PM set is mounted on either end of the table and machine ground, respectively, providing assistive repelling forces (F_{PM1} and F_{PM2}). The PM positions (x_{PM1} and x_{PM2}) are

controlled with servo mechanisms driven by rotary motors (RMs) to provide optimal magnetic gaps (d_1 and d_2) from the table position (x_T). The repelling forces of PMs exponentially increase as the gaps become smaller, conveniently providing smooth transitions from the no assist to the assist cases or vice versa. The length of table is denoted with a constant l_T . A displacement sensor (e.g. a linear encoder) is employed to measure the relative displacement between x_T and x_B , providing the stage displacement, x (not shown in Figure 1), to be FB controlled. Figure 2 shows the in-house built magnet assisted scanning stage prototype, which is used for system identification and simulation analysis of the control methods presented in this paper. Its performance specifications are summarized in Table 1 [1].

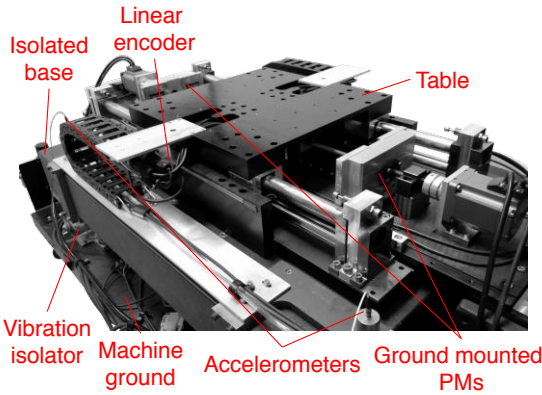


FIGURE 2. In-house built magnet assisted scanning stage prototype. Table mounted permanent magnets (PMs) are located under the table. Accelerometers measure the state of isolated machine base.

TABLE 1. Performance specifications of magnet assisted stage prototype reported in [1].

Stage travel	0.3 m
Max. acceleration	35 m/s ²
Table size (l_T)	0.36 m
Table mass (m_T)	13.8 kg
Base mass (m_B)	287 kg

MODELING AND SYSTEM IDENTIFICATION OF PNEUMATIC VIBRATION ISOLATORS

In this section, the dynamics of the passive pneumatic vibration isolators is modeled and its parameters are identified using the prototype stage shown in Figure 2.

Modeling of pneumatic vibration isolators

With absence of the PMs (i.e. with PMs positioned far away from the table), the model of horizontal stage dynamics considered in this paper reduces to Figure 3. Its dynamics is described by a mass-spring-damper system with internal linear motor force (F_{LM}), where m , k and b respectively denote masses, (equivalent) springs and (equivalent) dampers.

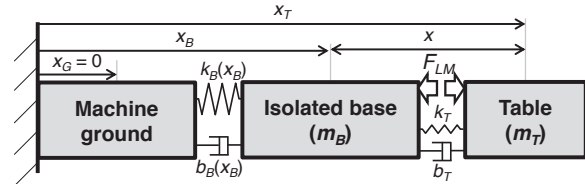


FIGURE 3. Model of scanning stage with vibration isolators.

The ground vibration (x_G) is assumed to be zero for simplicity, because ground vibration is small compared to x_B and x_T in well controlled environments such as semiconductor fabrication facilities. The transfer function (G_B) describing the input force (F_{LM}) and output displacement (x_B) relationship of the system model is given by

$$G_B s = \frac{-m_T s^2}{m_T m_B s^4 + \alpha s^3 + \beta s^2 + b_T k_B s}; \quad (1)$$

$$\alpha = m_B b_T + m_T b_B + m_T b_T$$

$$\beta = m_T k_B + b_B b_T$$

where s is the Laplace variable and the subscripts B and T are used to denote parameters pertaining to the vibration isolator and the table's guiding mechanism, respectively. In deriving Eq. (1), k_T is set to zero, since guiding mechanisms commonly used in precision scanning stages like the prototype stage shown in Figure 2 have negligible stiffness. The vibration isolator parameters (k_B and b_B) are assumed to be functions of x_B [6 – 9].

System identification of vibration isolators

The horizontal dynamics of vibration isolation system is identified using swept sine frequency response functions (FRFs). Experimental results with constant input force amplitude sine sweep FRFs are shown in Figure 4. The results support the nonlinearities reported in the literature; the input force amplitudes are varied from 15 N to 177 N. The nonlinearities shown here mainly

come from the dynamics of rubber diaphragms [6].

In order to identify the x_B dependent k_B and b_B , constant output (x_B) amplitudes are used to reconstruct FRFs based on the measured data shown in Figure 4 [9, 10]. In this reconstruction, the input force amplitudes are varied to achieve the same output amplitude level while input frequencies are swept. Then, Eq. (1) is used to fit the reconstructed FRFs with the respective x_B amplitudes using the known model parameters (m_T , m_B and b_T). Figure 5 shows the identified k_B and b_B as functions of x_B using 14 output amplitudes.

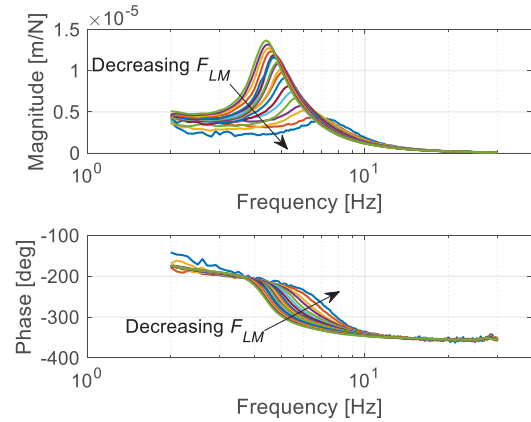


FIGURE 4. Constant input force frequency response functions of the isolated base. Linear motor force (F_{LM}) and base displacement (x_B) are the system input and output, respectively. The amplitudes of F_{LM} vary from 15 N to 177 N.

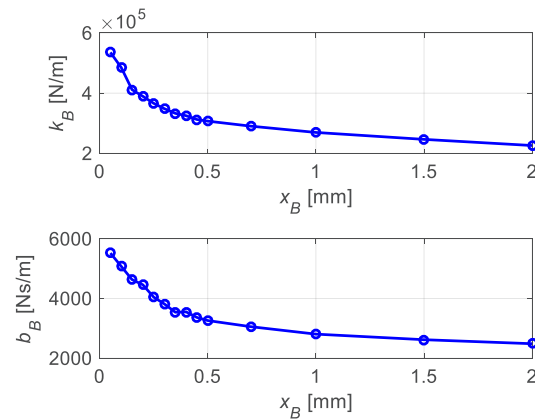


FIGURE 5. Identified vibration isolator stiffness (k_B) and damping (b_B) coefficients as functions of base displacement (x_B).

Figure 6 shows the time domain responses to two successive step motions of the prototype

stage and the identified system, respectively. The acceleration of reference motion command (a_{ref}) is laid out in the figure to show the timing of reaction F_{LM} applied to the base. Although the modeled response shows some discrepancies compared to the measured response in the time domain, the oscillation frequency and decay rate are similar to each other.

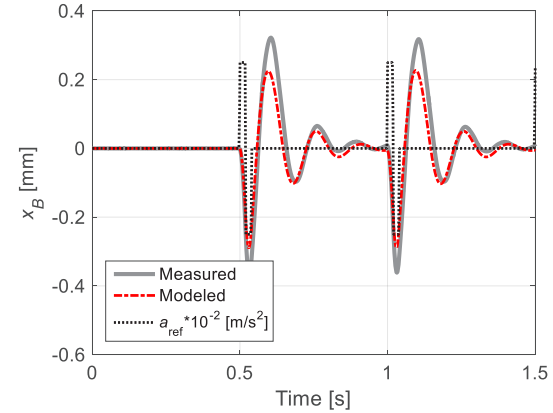


FIGURE 6. Modeled and measured responses of the prototype stage to piecewise constant acceleration commands.

COMPARISON OF OPTIMAL CONTROL STRATEGIES FOR PM ASSIST DEVICES

With the PM assist devices onboard, the task of controlling the PM assist devices (i.e. F_{PM1} and F_{PM2}) boils down to optimizing the trajectories of x_{PM1} and x_{PM2} . In this paper, two objective functions for minimizing base vibration are compared. The first objective focuses on minimizing F_{LM} as an indirect way of minimizing x_B , while the second objective minimizes x_B , predicted using the modeled isolation system dynamics.

Optimal control of PMs – no consideration of system dynamics

The first objective function (J_F) is given as

$$J_F = 1 - w \mathbf{F}_{LM}^T \mathbf{F}_{LM} + w \mathbf{F}_{RM1}^T \mathbf{F}_{RM1} + \mathbf{F}_{RM2}^T \mathbf{F}_{RM2} \quad (2)$$

where $\mathbf{F}_{LM}$, $\mathbf{F}_{RM1}$ and $\mathbf{F}_{RM2}$ are actuation effort vectors in discrete time domain for the linear motor (LM) and two rotary motors (RMs) for PM based assist devices, respectively. A user defined weight (w) can be selected very small (i.e. $w \ll 0.5$) because the costs associated with the RMs are lower than the LM. Note that this objective function J_F is identical to the objective function J_{contr} reported in the prior work by the

authors [1]. The actuator force vectors ($\mathbf{F}_{LM}$, $\mathbf{F}_{RM1}$ and $\mathbf{F}_{RM2}$) are computed with the discrete time sampled trajectories of $\mathbf{x}_{ref}$, $\mathbf{x}_{PM1}$ and $\mathbf{x}_{PM2}$. The dynamics of the isolation system is not considered using J_F . The actuator forces are computed using

$$\mathbf{F}_{LM} = m_T \mathbf{a}_{ref} + b_T \mathbf{v}_{ref} + \mathbf{F}_{PM1} - \mathbf{F}_{PM2} \quad (3)$$

$$\mathbf{F}_{RM1} = m_{PM1} \mathbf{a}_{PM1} + b_{PM1} \mathbf{v}_{PM1} - \mathbf{F}_{PM1} \quad (4)$$

$$\mathbf{F}_{RM2} = m_{PM2} \mathbf{a}_{PM2} + b_{PM2} \mathbf{v}_{PM2} + \mathbf{F}_{PM2} \quad (5)$$

where $\mathbf{v}$ and $\mathbf{a}$ are used to denote the 1st and 2nd discrete time derivatives the trajectories (i.e. velocity and acceleration of the reference), respectively. The parameters m_{PM1} , m_{PM2} , b_{PM1} and b_{PM2} respectively represent the mass of each PM (and servo mechanisms attached to it) and viscous damping of each servo guiding mechanism. The repelling PM forces ($\mathbf{F}_{PM1}$ and $\mathbf{F}_{PM2}$) are solved using the gaps between each PM and the table (see Figure 1). Their relationships are described by exponential functions of the gaps [1]. Details on the optimization solving and constraints enforcement (e.g. kinematic limits of RM actuators and ground vibration transmissibility) are presented in [1].

Optimal control of PMs – consideration of system dynamics

An objective function (J_B) considering the system (i.e. isolated base) dynamics is given as

$$J_B = \mathbf{x}_{B,sim}^T \mathbf{x}_{B,sim} + r \mathbf{F}_{RM1}^T \mathbf{F}_{RM1} + \mathbf{F}_{RM2}^T \mathbf{F}_{RM2} \quad (6)$$

where $\mathbf{x}_{B,sim}$ is the system response to the actuator input ($\mathbf{F}_{LM}$). A user defined weight (r) can be selected very small, because the rationale behind minimizing J_B is the maximum reduction of residual vibrations. A small non-zero r ensures that the RM actuators do not waste their efforts (i.e. motor currents; thus RM heating) doing unnecessary work.

SIMULATION ANALYSIS

The effectiveness of the two objective functions reducing $\mathbf{x}_B$ is evaluated using a simulation study based on the identified nonlinear stage dynamics including the FB controller and motor dynamics [1, 11]. A limited jerk (s-curve) reference trajectory ($\mathbf{x}_{ref}$) shown in Figure 6, which represents silicon wafer scanning trajectories commonly employed in industry, is used. The maximum speed and acceleration of

the reference trajectory are 0.5 m/s and 25 m/s², respectively.

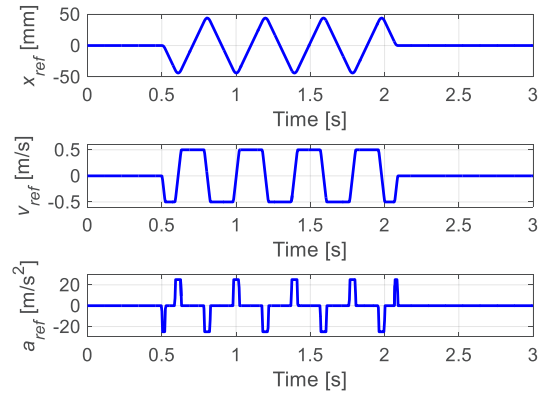


FIGURE 7. Reference trajectory ($\mathbf{x}_{ref}$) and its time derivatives. Reference velocity and acceleration are denoted with $\mathbf{v}_{ref}$ and $\mathbf{a}_{ref}$, respectively.

Effect of weight on optimization solutions

The effects of weights (w and r) used in the bi-objective optimizations on $\mathbf{F}_{LM}$ and $\mathbf{x}_B$ during scanning are evaluated as shown in Figure 8.

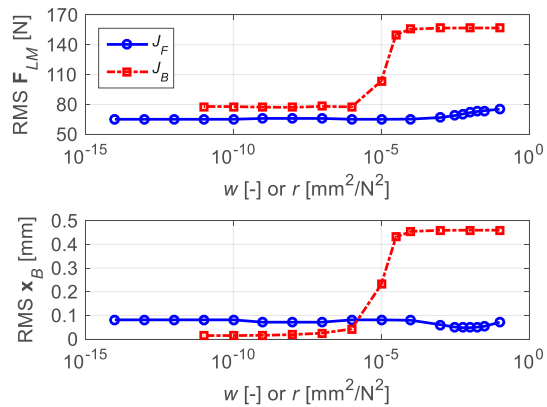


FIGURE 8. Effect of weight (w and r) on motor force ($\mathbf{F}_{LM}$) and residual vibration ($\mathbf{x}_B$) in optimization solutions using J_F and J_B .

Overall, lower weights used in optimization tend to produce smaller RMS $\mathbf{F}_{LM}$. However, due to the system dynamics being neglected in solving the optimization with J_F , the minimum RMS $\mathbf{F}_{LM}$ is not achieved with the minimum w . Moreover, lower RMS $\mathbf{F}_{LM}$ does not always guarantee lower residual vibration (i.e. RMS $\mathbf{x}_B$) due to the system dynamics as shown in Figure 8. Compared to the case with $w = 10^{-3}$, the larger $w = 10^{-2.5}$ produces a smaller RMS $\mathbf{x}_B$, even if its RMS $\mathbf{F}_{LM}$ is larger. For the case with J_B , RMS $\mathbf{x}_B$ is monotonically decreasing with reduced r . For

each objective function case, residual vibration minimizing weights ($w = 10^{-2}$ and $r = 10^{-12}$) are selected to solve the optimization problems.

Comparisons of two objective functions

Figure 9 shows the time domain responses of the two optimal solutions using simulation, while Figure 10 shows the cumulative distribution of base displacement over the 3 s long operation of the stage. The optimization solution with J_B produces smaller base displacement as shown in both the time domain and cumulative distribution figures.

Figure 11 shows the frequency spectra of the isolation system input (F_{LM}) and output (x_B) for the two optimization solutions using discrete Fourier transform (DFT) of the signals. The J_F minimizing solution produces the smaller input force (i.e. RMS F_{LM}), which is 72 N vs. 78 N of the J_B case. However, due to the system dynamics, the output responses to input are amplified in the low frequency range (about 5 – 7 Hz), where the system resonance occurs as illustrated with the FRFs in Figure 4. The J_B minimizing solution successfully avoids the vibration amplification near the resonance; thus the vibration is effectively reduced.

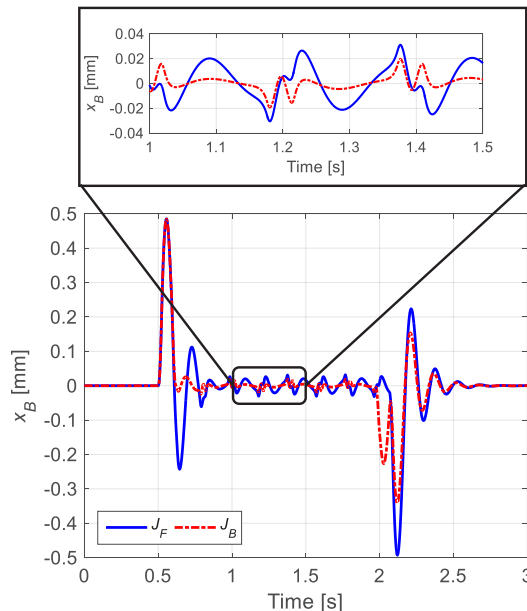


FIGURE 9. Simulated base displacement when the stage processes the high throughput motion shown in Figure 7. Optimal control of PMs using J_F and J_B objectives are shown to reduce residual vibration of base.

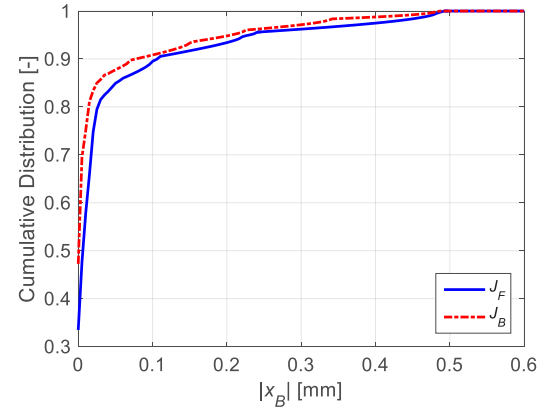


FIGURE 10. Cumulative distributions of x_B magnitudes. Two objective functions (J_F and J_B) are used to generate PM assist device trajectories.

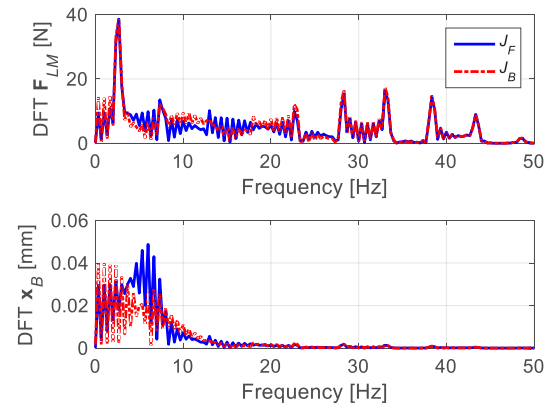


FIGURE 11. Frequency spectra of isolation system input (F_{LM}) and output (x_B) of optimization solutions using the two objective functions (J_F and J_B).

CONCLUSIONS AND FUTURE WORK

The residual vibration of passive pneumatic isolators, created by the force disturbance of linear motors in precision scanning stages, is mitigated using PM assist devices considering the dynamics of the system. Compared to the input force minimization objective without consideration of the system dynamics, which is the prior work of the authors, the inclusion of system dynamics in the objective function provides a more effective PM based assist device trajectory solution. Future work will investigate computationally efficient solution processes for optimal assist device control, since the optimization considering the dynamics can be computationally expensive. In addition, the authors will experimentally validate the

effectiveness of proposed control objective using the prototype stage.

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AN IDENTIFICATION METHOD OF FORCE RIPPLE IN PERMANENT MAGNET SYNCHRONOUS LINEAR MOTORS BASED ON THE COMBINATION OF RELAY FEEDBACK AND GENETIC ALGORITHM

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INTRODUCTION

PMSLMs are widely used in industrial production[1,2]. In the meanwhile, PMSLMs have overcome difficulties such as low efficiency, large volume and low accuracy. As a compromise, any external disturbance will directly act on the output terminal of PMSLMs, which results in low tracking accuracy and significant nonlinear characteristics. As a prominent case, magnetic field distortion may lead to force ripple that includes cogging and reluctance forces, both of which show a periodic relationship with respect to the displacement of the translator and the stator[3].

To address the problem, this paper mainly focuses on a novel identification method based on the optimized relay apparatus and a common numerical solution method. First of all, the mathematical model of the servo-mechanical system was established. Subsequently, in order to excite oscillations in the feedback loop, the optimized relay apparatus were described. Afterwards, the details of the identification procedure including the basic relay components, requirements for exciting oscillation and the numerical calculation method were elaborated comprehensively. Moreover, Simulations were conducted to investigate how oscillations were influenced by different relay gains. Laying on the foundation of the model of the friction and force ripple, a feed forward control scheme was implemented and compared with a traditional control strategy. Finally, conclusions were drawn.

SERVO-MECHANISM DYNAMICS

Considering a general model of PMSLMs, the servo-mechanical system which combines the mechanical dynamics and electrical dynamics can be represented by nonlinear mathematical models

$$u(t) = K_e \dot{x} + Ri(t) + L \frac{di(t)}{dt} \quad (1)$$

$$f(t) = K_f i(t) \quad (2)$$

$$f(t) = m\ddot{x}(t) + f_r + f_f + f_\Delta \quad (3)$$

where $u(t)$ is the armature voltage while $i(t)$ is the thrust current, and $f(t)$ is the developed force; $x(t)$ refers to the motor position; f_r denotes the ripple force of the linear motor; f_f is the friction force and f_Δ represents the other uncertain disturbances. Other parameters are described in Table 1.

Table 1. Linear motor parameters

Parameters	Physical	Units
m	Mass of	kg
K_e	Back EMF	V/(m/s)
K_f	Force	N/A
R	Resistance	Ω
L	Armature	mh

The friction force f_f can be considered as a combination of Coulomb and viscous friction. The model of the friction can be expressed according to as

$$f_f = f_c \operatorname{sgn}(\dot{x}) + f_v \dot{x} \quad (4)$$

where f_c is the minimal level of Coulomb friction and f_v is viscous friction coefficient.

In view of the fact that force ripple is associated with the displacement between the stator and the translator of PMSLMs, the force ripple f_r can be described by a standard sinusoidal function with phase shift θ whose single dominant spatial frequency can be determined by experiment:

$$\begin{aligned} f_r &= C \sin(\Omega x + \theta) \\ &= C_1 \cos(\Omega x) + C_2 \sin(\Omega x) \end{aligned} \quad (5)$$

According to the (2)-(5), the following simplified equivalent model is obtained

$$m\ddot{x} + f_v \dot{x} = F' \quad (6)$$

$$F' = K_f i(t) - f_c \operatorname{sgn}(\dot{x}) - C_1 \cos(\Omega x) - C_2 \sin(\Omega x) \quad (7)$$

Thereafter, transfer function of the linear dynamic of the servo-mechanical system can be expressed as follow:

$$G(s) = \frac{X(s)}{\tilde{F}(s)} = \frac{1}{ms^2 + f_v s} = \frac{b}{s+a} \cdot \frac{1}{s} \quad (8)$$

Set $a = f_v/m$, $b = 1/m$, Then, terms m , f_c , f_f , C_1 and C_2 are the parameters required for identifying in this paper.

IDENTIFICATION BASED ON RELAY FEEDBACK AND GENETIC ALGORITHM Requirements For Exciting The Oscillation Of The Servo-Mechanism

In order to excite the oscillation of the nonlinear system, a hysteresis relay component along with dead zone and an ideal relay element were implemented in the feedback loop. The standard ideal relay element can be expressed as

$$I_{ideal} = \begin{cases} D, & \text{if } e > 0 \\ -D, & \text{if } e < 0 \end{cases} \quad (9)$$

while the hysteresis relay component along with dead zone can be illustrated as shown in Figure 1.

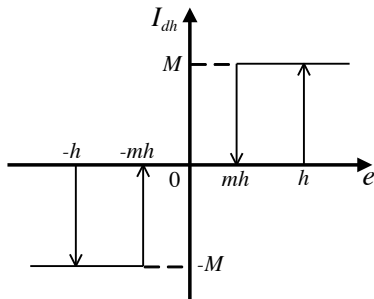


FIGURE 1. Intentional relay apparatus used for identification.

It can be defined as follow:

$$I_{dh} = \begin{cases} M, & \text{if } e > h, \\ & \text{or } (mh < e < h \text{ and } i(t_-) = M) \\ 0, & \text{other} \\ -M & \text{if } e < -h, \\ & \text{or } (-h < e < -mh \text{ and } i(t_-) = -M) \end{cases} \quad (10)$$

In this way, combined with the intentional relay components, the control system block diagram of the servo-mechanism can be built as shown in Figure 2. Afterwards, the identified parameters can be determined according to the characteristics of the oscillation excited by the reasonable relay gains in the servo-mechanism system.

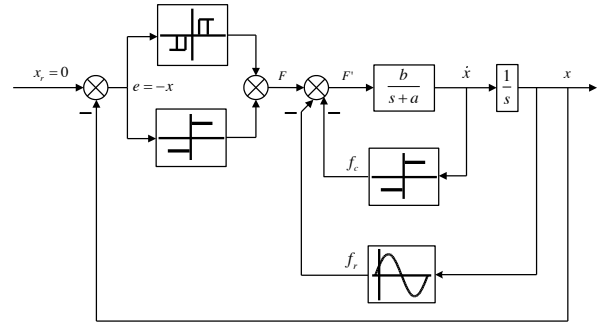


FIGURE 2. Control system block diagram for identification.

Solving Parameters By Using Genetic Algorithm

The unknown parameters can be obtained according to the relay gains, oscillation and the model of the servo-mechanism. The objective function of the identification formula can be built as follow:

$$J = \min \|I - B\Omega\|_2^2 \quad (11)$$

where $I = \operatorname{diag}(1, 1, 1)$, $B = \operatorname{diag}(a_0, b_0, \omega_0)$ whose a_0 , b_0 and ω_0 denotes the amplitude, offset and frequency of the standard outputs while Ω represents the amplitude, offset and frequency of the extracted outputs. Then, the genetic algorithm was employed to solve the identified parameters as the flow chart shown in Figure 3.

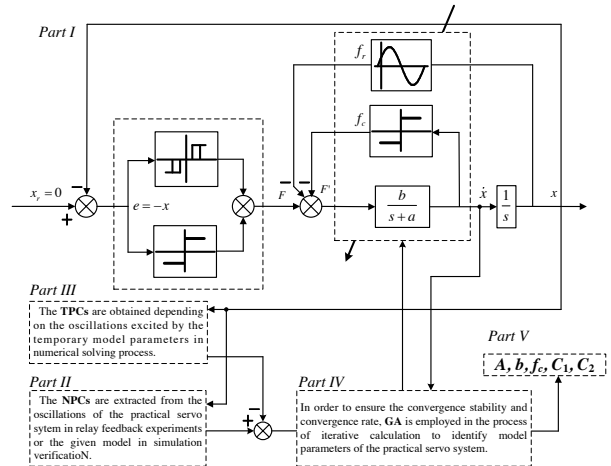


FIGURE 3. Parameter identification flow chart.

Simulation verification

In this section, we will utilize the Matlab/Simulink to simulate the nonlinear system. To verify the feasibility and identification accuracy, basic parameters of the servo-mechanism are

assumed to compare with identification results. Dynamics of the linear portion, Coulomb friction factor, amplitude and single dominant spatial frequency of the force ripple are assumed as follows

$$a = 4, \quad b = 40, \quad C = 1, \quad f = 0.4 \text{ and } \Omega = 0.2\pi.$$

For all the following simulations, the sampling time is configured as 0.1ms.

Various Limit Cycle In Different Relay Gains

Basically, when the parameter m of the intentional relay apparatus is set as 1, the intentional relay apparatus will degrade to a standard relay with dead zone. Subsequently, setting the phase of force ripples as zero, oscillations can't be found according to the method of describe function[4]. In other words, negative oscillation frequency is not possible to be found when the relay gains are implemented on the actual servo system.

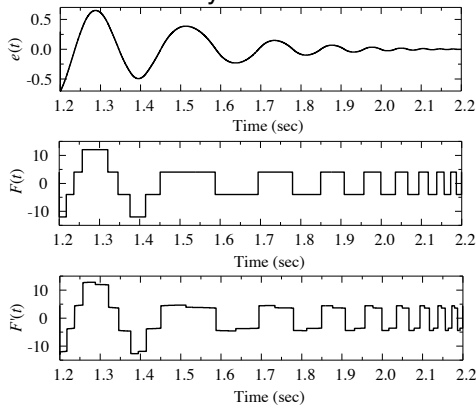


FIGURE 4. Input $e(t)$, output $F(t)$ and actual control signal $F'(t)$ with $M = 8$, $D = 4$, $h = 0.4$, $m = 1$, and $\theta = 0$

In the first scenario as shown in Figure 4, the sustained limit cycle does not exist as previous demonstration.

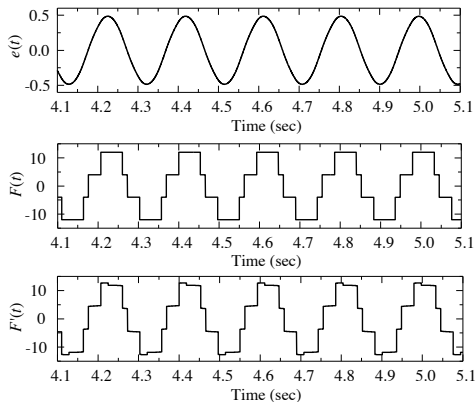


FIGURE 5. Input $e(t)$, output $F(t)$ and actual control signal $F'(t)$ with $M = 8$, $D = 4$, $h = 0.4$, $m = 0.5$, and $\theta = 0$.

As the parameter m reduces from 1 to 0.5, the symmetric oscillation appears as shown in Fig.5.

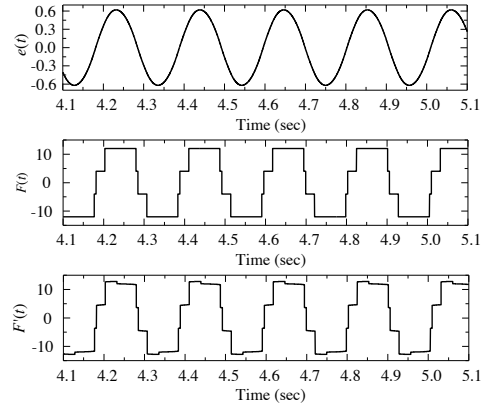


FIGURE 6. Input $e(t)$, output $F(t)$ and actual control signal $F'(t)$ with $M = 8$, $D = 4$, $h = 0.4$, $m = 0.25$, and $\theta = 0$

According to the above-mentioned analysis, since the parameter θ is set as zero, we realize that the oscillations can be symmetric without existence of biased term caused by the even nonlinearity. Furthermore, as the value of m decreases, we discover that the amplitude increases while the frequency of oscillations decreases as shown in Fig.6. Based on the above analysis, the correctness of the formula (24) can be confirmed basically.

Model Identification

In this paper, applying optimized relay elements to the servo-mechanical system, sustained oscillations can be excited. Furthermore, for the sake of extracting the frequency and amplitude in the oscillations, DFT is utilized to several complete cycles of $e(t)$. To verify the accuracy and practicability of the identification method, we assume that the model of the servo-mechanical system is known in advance. With different relay gains, ideal exciter, actual control signal and oscillation are exhibited in Figure 7 and Figure 8.

The numerical calculation method as shown in Figure 3 was employed to solve the identified parameters in the servo-mechanical system. The model parameters are identified with the following initial guess of $a = 2$, $b = 35$, $C_1 = 0.1$, $C_2 = 0.1$ and $f = 0.1$ and with the variation range of $[0, 8]$, $[30, 50]$, $[0, 2]$, $[0, 2]$ and $[0, 2]$, respectively.

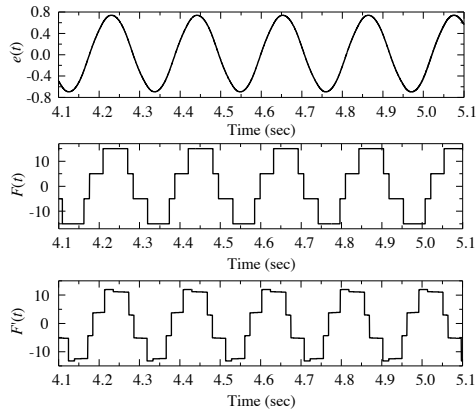


FIGURE 7. Input and output of the optimized relay apparatus and actual control signal $F'(t)$ with $M = 10$, $D = 5$, $h = 0.6$, $m = 0.5$, and $\theta = \pi/6$

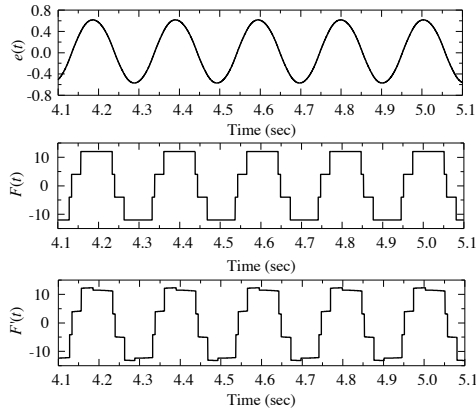


FIGURE 8. Input and output of the optimized relay apparatus and actual control signal $F'(t)$ with $M = 8$, $D = 4$, $h = 0.4$, $m = 0.3$, and $\theta = \pi/6$

Table 2 shows the summary of simulation results based on the numerical calculation method. Thus it can be seen, the estimation error can stay within 8% which reveals the effectiveness of the identification method.

Table 2. Summary of simulation results

Parameters	Physical	Units	Error
a	4.0	4.0420	1.05
b	40.0	40.3416	0.854
C_1	0.5000	0.4628	-7.44
C_2	0.8660	0.8594	-0.76
f	0.4	0.3928	-1.68

A FEEDFORWARD CONTROL STRATEGY BASED ON THE IDENTIFICATION RESULTS

In this part, to verify the identification accuracy of the friction and force ripple in the servo-mechanical system, tracking error will be compared under two different control strategies. Figure 9.(b) shows the tracking performance of a traditional PID controller, while Figure 9.(c) exhibits the tracking error of a traditional PID controller with a feed forward compensator using

the obtained model of friction and force ripple. The desired moving profile is set as $x_d = 0.01 \sin(0.5\pi t)$ (m). From Figure 9.(b), the maximum tracking error is around 333 μm without the nonlinear feed forward compensator. Figure 9.(c) has illustrated the efficiency of the compensator since the maximum tracking error decreases to 113 μm . Evidently, with the feed forward compensator, the root mean square error (RMSE) of the tracking error can be drastically reduced from 187.7 μm to around 57.1 μm . Obviously, results from the experiment have verified the practicability of the proposed method.

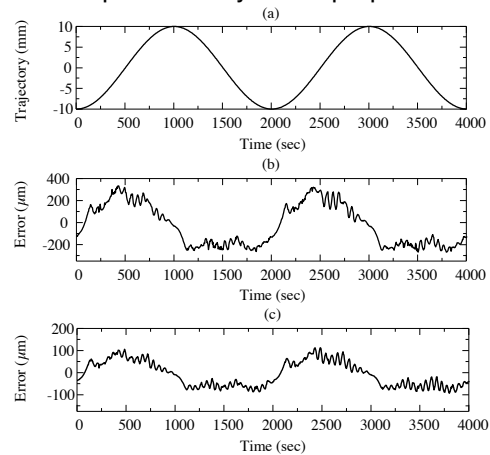


FIGURE 9. Experiment results. (a) Desired profile. (b) Without nonlinear feed forward compensator. (c) With nonlinear feed forward compensator.

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UNCERTAINTY QUANTIFICATION IN DIMENSIONAL MEASUREMENTS BY COMPUTED TOMOGRAPHY DUE TO UNCERTAINTY IN DATA ACQUISITION GEOMETRICAL PARAMETERS

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INTRODUCTION

X-ray computed tomography (CT) is an imaging technique that provides a slice-wise volumetric reconstruction of the imaged part. Dimensional measurements can be performed on the reconstructed volume, making CT an attractive solution for non-destructive dimensional quality control, for example, of complex parts with inner geometries. The scanned object is reconstructed into a three-dimensional attenuation image by applying a tomographic reconstruction algorithm to the acquired 2D projections; the Feldkamp-Davis-Kress (FDK) algorithm is commonly used for reconstruction in cone-beam CT systems. Surfaces are segmented from the reconstructed volume and features are measured by fitting of geometrical primitives (spheres, planes, cylinders, etc.).

Dimensional measurements made by CT are sensitive to geometrical misalignments of the measuring instrument [1]. We define misalignments as discrepancies between the actual geometrical parameters of the test system and the geometrical parameters used in the back projection step of the tomographic reconstruction. Calibration procedures provide users with the measurement of the CT geometry. However, uncertainty in the measured geometrical parameters propagate to uncertainty in the reconstructed volume and, consequently, to uncertainty in the dimensional measurements. Quantification and propagation of such sources of uncertainty must be considered to provide a complete uncertainty analysis and to establish

confidence intervals for dimensional measurements obtained from volumetric data.

In this paper we describe a new method to estimate uncertainty in CT dimensional measurements due to the detector angular misalignments by means of Monte Carlo simulation and a dedicated geometrical error model. The model consists of first applying of forward projection to coordinates corresponding to surface points extracted from an object's analytical model to determine the pixel coordinate of their projection on the misaligned detector. Then, projected points are back-projected with assumption of perfectly aligned detector. As a result reprojection error for each surface point is estimated. Detector alignment is described by Tait-Bryan rotations and uncertainty in angles is parameterized as a uniform distribution around their measured values. The proposed approach provides computational benefits since explicit evaluation of uncertainty in the surface coordinates avoids the expensive computational burden of the CT reconstruction and segmentation steps. The efficacy of the proposed approach was confirmed on simulated data in the presence of individual uncertainty contributors.

DIMENSIONAL MEASUREMENTS BY COMPUTED TOMOGRAPHY

In cone-beam CT the object is exposed to a cone-shaped X-ray beam, which diverges from the source, through the measurement volume, and onto a two-dimensional flat panel detector. The cone-beam can be parameterized by a

series of trajectories from each detector pixel to the X-ray source. The X-rays are attenuated depending on the attenuating properties of the measured object along the corresponding trajectory. The intensity of X-rays along a given path is measured by the corresponding detector pixel. Radiographs are acquired from different angular positions of the measured object and stacked into acquisition dataset. Accurate back projection of measured X-ray intensities therefore depends on accurate knowledge of the CT geometrical parameters, i.e. the position and orientation of X-ray source, object, and detector for each radiograph.

Most conventional industrial CT systems employ FDK reconstruction due to its computational benefits. The core of the algorithm is based on back-projection of the 2D radiographs along original rays and superposition of multiple radiographs, which makes reconstruction sensitive to errors in geometrical parameters. When paths of original rays are not accurately known, then the superposition of projections might not be consistent. For a given point $x = (x, y, z)$ in measurement volume rays corresponding to this point will not be back projected to x but will draw multiple intersections at erroneous positions, thereby resulting in multiple points with lower attenuation values scattered around x . That is, various artifacts can exist in the reconstructed volume, such as blurring, double-borders, visual artifacts, erroneous magnification, which will propagate to errors in segmented surface location and consequently to dimensional measurements. We show in figure 1 the cross sections of a reconstructed sphere from ideal (left) and misaligned detector (right). We refer to error in reconstructed surface location to as volumetric error.



FIGURE 1. (Left) reconstruction of middle slice of the sphere under ideal CT geometry. (Right) Double-borders due to detector misalignment.

CT Geometry

The geometry of a CT system is defined in terms of a Cartesian coordinate system [1]. The following description is supported by the diagram in figure 2. A Cartesian coordinate system is defined and its origin is coincident with the isocenter $O = (0, 0, 0)$. The Z axis coincides with the magnification axis, which is defined as the line containing the X-ray source and normal to the ideal detector. The Z axis is positive towards the X-ray source. The Y axis is coincident with the rotation axis and is antiparallel to gravity. The X axis is given by the right-hand-rule. The origin of the coordinate system, namely the isocenter, is located at the intersection of the ideal magnification and rotation axes. The coordinates of the X-ray source are given by $s = (0, 0, SRD)$, where SRD is the distance from X-ray source to the rotation axis along the magnification axis. For simplicity, the X-ray source is described as an infinitesimally small point. The orientation and position of the detector plane is defined by three orthogonal unit vectors e_u, e_v, e_n and principal point $p_0 = (0, 0, SRD - SDD)$, respectively. SDD is the distance from X-ray source to the detector along the magnification axis. Vectors e_u, e_v correspond to the two orthogonal axes in the plane of the detector (see figure 2).

A point in the measurement volume is given by $x = (x, y, z)$. During acquisition, the X-ray source and detector are fixed, while the object is rotated around the Y axis in angular increments of $\Delta\alpha$. For a given rotation angle α , the rotated volumetric coordinate x_α is given by

$$x_\alpha = R_\alpha x, R_\alpha = \begin{bmatrix} \cos \alpha & 0 & \sin \alpha \\ 0 & 1 & 0 \\ -\sin \alpha & 0 & \cos \alpha \end{bmatrix} \quad (1)$$

In this work, the detector orientation is parameterized by three extrinsic rotations θ, ϕ, η around the X (tilt), Y (slant) and Z (skew) axes, respectively, with the center of rotation at the detector principal point p_0 , i.e. the intersection of magnification axis and detector (figure 3). The orientation of the misaligned detector is defined by its normal e'_n , which is a function of the three rotation matrices R_θ, R_ϕ, R_η and the detector ideal normal e_n :

$$e'_n = R_\phi R_\eta R_\theta e_n \quad (2)$$

The rotated detector coordinate axes e'_u, e'_v in the presence of misalignment are given by:

$$\mathbf{e}'_u = R_\phi R_\eta R_\theta \mathbf{e}_u \quad (3)$$

$$\mathbf{e}'_v = R_\phi R_\eta R_\theta \mathbf{e}_v \quad (4)$$

See reference [2] for further detail on rotation conventions.

METHOD

Test Object

While it is possible to extend the method to complex geometries and arrangement of objects in the measurement volume, in this paper we focus on the implementation of our method to spheres to lay the ground-work for more complex objects. The edge of a projected sphere with radius R and center point $\mathbf{c} = (c_x, c_y, c_z)$ can be approximated analytically as an ellipse. The projection edge is generated by the edge of the sphere traced by circle D (figure 4). Circle D corresponds to the edge of the sphere for which X-ray paths are tangential. The collection of lines along circle D form a cone with vertex at $\mathbf{s}$. Circle D can be described as the intersection of the sphere surface and a plane perpendicular to the

vector $\mathbf{n}$ from source $\mathbf{s}$ to center of the sphere $\mathbf{c}$. Analytically, circle D is defined by the intersection of sphere and the plane parameterized by normal vector $\mathbf{n}$ and point $\mathbf{p}$. The parametric equation of the circle D is given by equation (5).

$$\mathbf{D}(t) = r \cos t \mathbf{w} + r \sin t (\mathbf{n} \times \mathbf{w}) + \mathbf{p} \quad (5)$$

Where r is radius of circle D, $\mathbf{w}$ is any vector perpendicular to $\mathbf{n}$, $(\mathbf{n} \times \mathbf{w})$ denotes cross-product of vectors $\mathbf{n}$ and $\mathbf{w}$. Point $\mathbf{p}$ is given by:

$$\mathbf{p} = \mathbf{s} + \mathbf{n} \left(d - R \cos \left(\sin^{-1} \frac{\sqrt{d^2 - R^2}}{d} \right) \right) \quad (6)$$

Where d is the distance from $\mathbf{s}$ to $\mathbf{c}$.

The radius r is given by:

$$r = R \sqrt{1 - \cos^2 \left(\sin^{-1} \frac{\sqrt{d^2 - R^2}}{d} \right)} \quad (7)$$

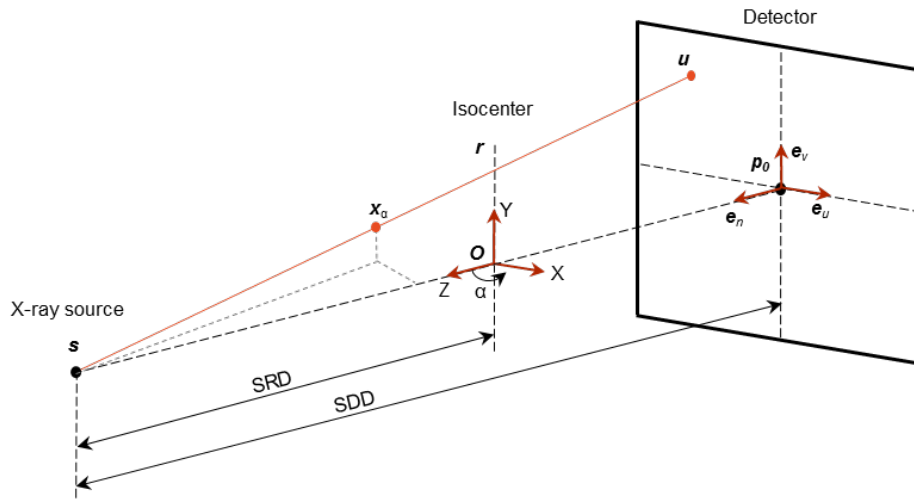


FIGURE 2. Cone-beam CT geometry

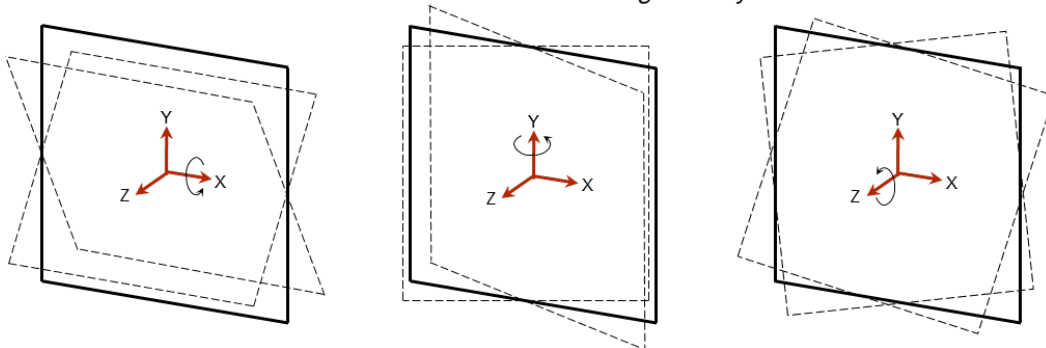


FIGURE 3. Angular misalignments of the flat-panel detector are parameterized as three rotation angles [2]. Left: tilt – rotation around X axis. Center: slant – rotation around Y axis. Right: skew – rotation around Z axis.

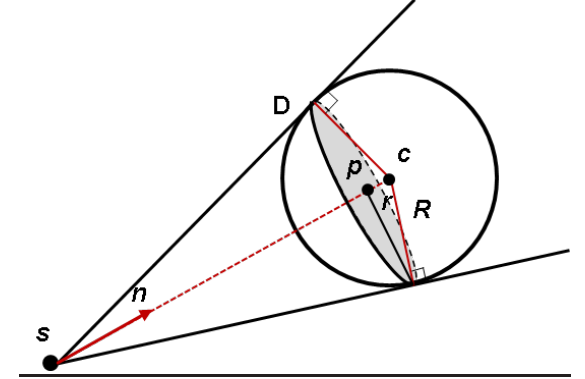


FIGURE 4. Cone tangent to the sphere. Tangent lines trace a circle D on the sphere surface.

Volumetric error model

Our volumetric error model requires the following as input parameters:

- CT system parameters, such as SRD , SDD , and detector size,
- sphere center $c = (c_x, c_y, c_z)$ and radius R ,
- detector angular misalignments θ , ϕ , and η
- number of object rotation positions m ,
- number of sampling points l along circle D.

Algorithm:

1. For each rotation position $\alpha_j = (j - 1) \frac{360^\circ}{m}$, $j = 1, \dots, m$, from equation (1) find position of center point c_α and sample circle D_α in $k_1, \dots, k_l$ points according to equations (5)-(7).
2. Find new detector orientation e'_n according to equation (2).
3. Trace the rays $q_1, \dots, q_l$ through sample points $k_{\alpha,1}, \dots, k_{\alpha,l}$.
4. Find intersection coordinates $f_1, \dots, f_l$ of rays $q_1, \dots, q_l$ with misaligned detector plane in XYZ coordinate frame.
5. Convert $f_1, \dots, f_l$ from XYZ coordinate frame to detector coordinate frame:

$$u' = (\langle f, e'_u \rangle, \langle f, e'_v \rangle), \quad (8)$$

Where $\langle \dots \rangle$ denotes scalar product and e'_u, e'_v are the in-plane coordinate axes of misaligned detector from equations (3) and (4).

6. Trace the rays $q'_1, \dots, q'_l$ through $u'_1, \dots, u'_l$ back to the source and define new circle D'_α sampling points $k'_{\alpha,1}, \dots, k'_{\alpha,l}$.
7. Rotate sampling points $k'_{\alpha,1}, \dots, k'_{\alpha,l}$ to zero rotation position

$$k'_{j,1}, \dots, k'_{j,l} = R_\alpha^{-1} k'_{\alpha,1}, \dots, k'_{\alpha,l} \quad (9)$$

Backprojection of D'_α from multiple rotation positions in the presence of detector misalignments results in two overlapping point clouds as shown in figure 5, left. We separate point cloud into two components: external and internal, which are shown in blue and red, respectively. Comparison with reconstructed data from simulation in the presence of equivalent detector misalignments (figure 5, right) shows that external point cloud corresponds to outermost contour of the object with lower intensity value. The internal point cloud, on the other hand, corresponds to the higher intensity region in the center of the reconstructed sphere. Consequently, depending on the threshold value chosen for surface extraction, either external or internal point cloud can be used for uncertainty quantification.

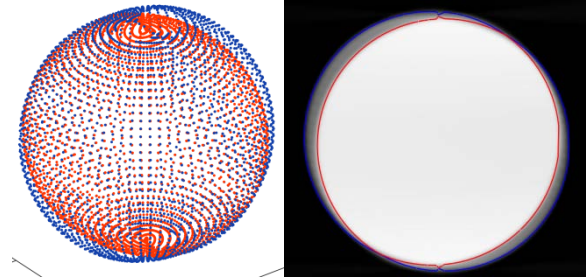


FIGURE 5. Results of proposed algorithm. Left: generated point cloud. Right: superposition of simulated data and generated point cloud.

Monte Carlo uncertainty quantification

Uncertainty in θ, ϕ, η is parameterized as zero-mean uniform probability distribution $\mathcal{U}(a, b)$. For each Monte Carlo run $i = 1, \dots, n$, we perform the following steps:

1. Sample detector orientation $\{\theta_i, \phi_i, \eta_i\}$ from uniform probability distribution.
2. Run algorithm and extract external surface.
3. Fit sphere to external point cloud to define sphere parameters $c'_i = (c'_x, c'_y, c'_z)$ and R' .

We use $c'_1, \dots, c'_n$ and $R'_1, \dots, R'_n$ to estimate the confidence interval of sphere center c and radius R .

RESULTS

The output of the proposed model is evaluated for the measurement of a cylindrical array of 5 mm diameter spheres (figure 6) in the presence of individual detector misalignments. One hundred and twenty five spheres are separated

into five layers along the cylindrical axis (figure 6, left). Each layer consists of twenty five spheres; the numbering of the top layer of spheres is shown in figure 6 (right) and numbering for subsequent layers continues in the same manner.

Figures 7-9 show preliminary comparison of the volumetric model output to simulated CT data in the presence of misalignments corresponding to boundary parameters a, b of uniform distribution $\mathcal{U}(a, b)$ for some representative results. The solid line corresponds to 95% confidence interval estimated by Monte Carlo simulation. Crosses correspond to the upper boundary parameter a , i.e. $\theta = 0.5$ and circles correspond to lower boundary parameter b , i.e. $\theta = -0.5$. Green, orange and red correspond to 25%, 50% and 75% threshold, respectively, used for segmentation.

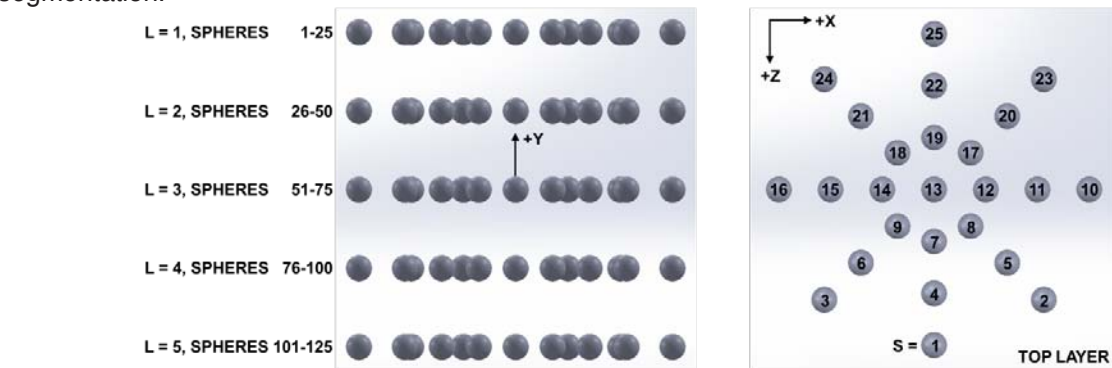


FIGURE 6. Cylindrical array of sphere. Left: lateral view. Right: top view.

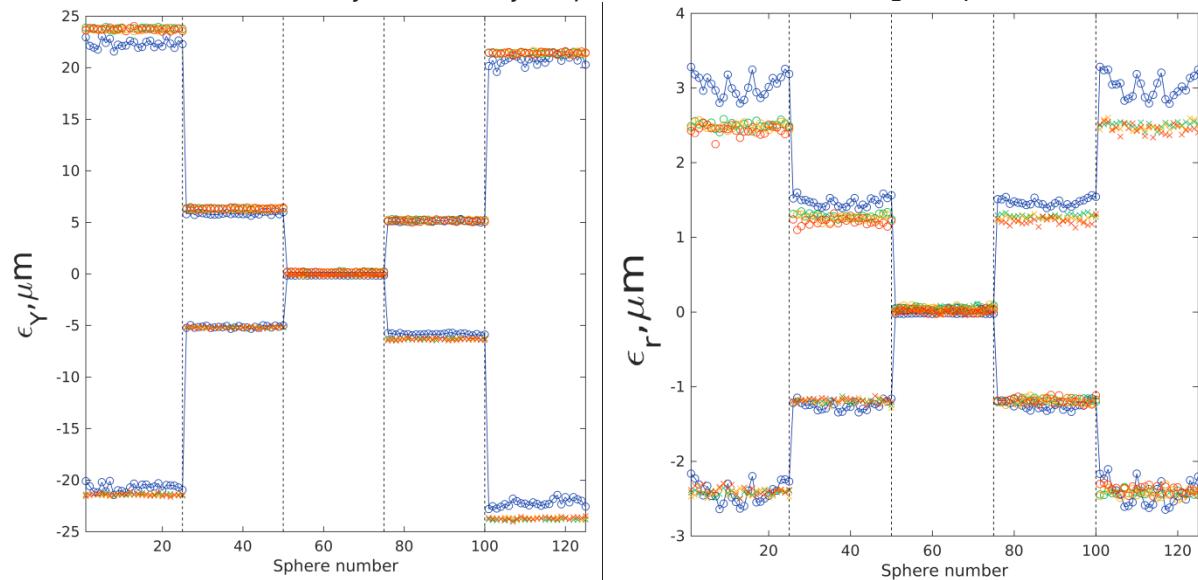


FIGURE 7. $\theta = \mathcal{U}(-0.5, 0.5)$. Left: estimated 95% confidence interval for c_y coordinate of center point c . Right: estimated 95% confidence interval for radius R

Results show that proposed model captures the systematic behavior observed in simulated data and can be used for estimating uncertainty due to the detector misalignments. Extension of the proposed method to other geometrical parameters and objects is topic of future investigation.

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- [2] Ferrucci M, Ametova E, et al. Evaluating the effects of detector angular misalignments on simulated computed tomography data Prec. Eng. 45 230-241

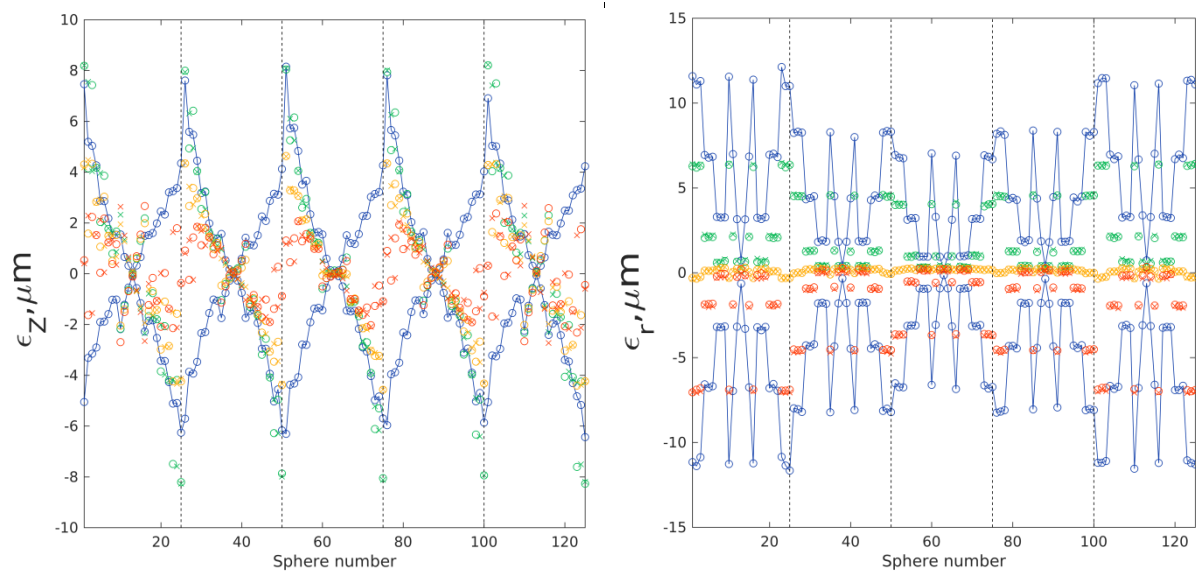


FIGURE 8. $\phi = \mathcal{U}(-1,1)$. Left: estimated 95% confidence interval for c_z coordinate of center point c . Right: estimated 95% confidence interval for radius R

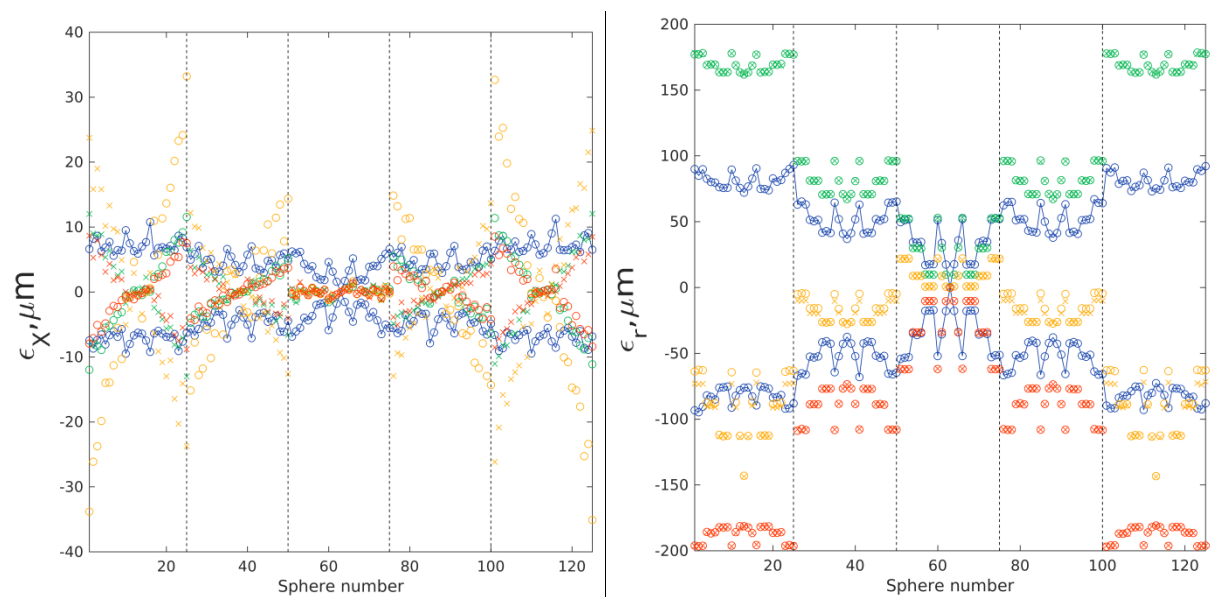


FIGURE 9. $\eta = \mathcal{U}(-0.5,0.5)$. Left: estimated 95% confidence interval for c_x coordinate of center point c . Right: estimated 95% confidence interval for radius R

OVERVIEW AND COMPARISON OF SPATIAL LIGHT MODULATOR CALIBRATION METHODS

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INTRODUCTION

Wavefront control is critical for advancing a wide array of optical systems, yet it is difficult to integrate a closed loop diagnostic subsystem. In this work, we explore different metrology approaches to identify fundamental electrical and physical errors in a wavefront-controlling spatial light modulator (SLM) and compensate for them. Three drop in diagnostic systems were created to analyze phase modulation by an SLM. A systematic approach to calibration is demonstrated. Our aim is to develop an optimal phase calibration apparatus which will enable significant advances in precision, repeatability, and predictability in optical systems such as digital holography and holographic optical tweezers [1, 2].

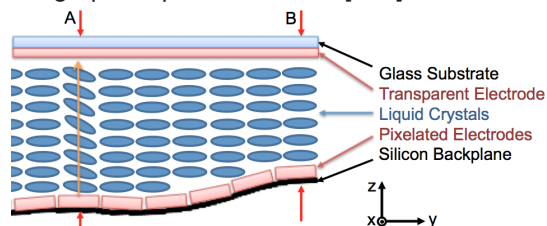


FIGURE 1. Framework of a parallel aligned nematic liquid crystal on silicon spatial light modulator. Layers include a glass substrate, continuous transparent electrode, liquid crystals, pixelated electrodes, and a silicon backplane. Thickness of SLM at A and B may vary due to fabrication imperfections, such as a non-flat silicon backplane. All pixels bear zero potential difference except the pixel at A.

In this study we use a parallel aligned nematic liquid crystal on silicon spatial light modulator (LCoS SLM) as illustrated in Figure 1 [3]. Cross

sectional layers of the SLM consist of a glass substrate, a transparent electrode, liquid crystals, pixelated electrodes and a reflective silicon backplane. As light passes through the liquid crystals, the phase is modulated proportionally to the effective refractive index of the liquid crystal, which can be altered by applying a voltage. Pixelated electrodes allow independent control of potential difference, providing the capability of spatially varying phase. However, fundamental electrical and physical hardware errors mandate calibration of the SLM.

The voltage applied to each pixel in the SLM is assigned through a look up table (LUT) programmed in the SLM control software. Eight-bit grey values displayed on the SLM control the voltages applied at each pixel. After observing a nonlinear relationship between the measured overall phase and applied grey value, it was determined that the LUT needed to be calibrated to recover a linear response. The gamma table and voltages that were encoded in the SLM are optimized by the manufacturer to yield a linear phase shift with grey value of range 2.1π .

Additionally, due to fabrication limitations, the silicon backplane and glass substrate of the SLM may not be flat nor parallel [4]. This contributes to the nonlinearity of the response, since the induced phase delay is proportional to thickness, illustrated in Figure 1. For this calibration, more precise measurements of spatially varying phase over the entire SLM surface needed to be made.

MEASUREMENT METHODS

Previous studies suggest three diverse techniques including interferometry, linear polarizers, and diffraction based measurements of phase delay to quantify variable phase error as a result of a non-flat backplane of the SLM, shown in Figure 1. Phase correction is dependent on both the physical backplane error and the nonlinear electrically induced birefringence inherent to liquid crystals. Interferometry based measurements lead to an analysis of both relative phase delay from active liquid crystals with respect to the off state (grey value of 0) and any curvature within the physical layers of the SLM. The linear polarizer technique described below measures the absolute phase change imposed by the liquid crystal, which can be measured for each pixel. Diffraction based measurements quantify an overall phase of the entire illuminated SLM area relative to an SLM with no applied voltage. Our aim is to perform a rigorous comparison of the techniques and quantify the reliability and repeatability of the processes aiming to create an *in situ* calibration method.

The SLM used to evaluate all of these techniques is a Holoeye-Pluto SLM (HES 6010 VIS) set to a 5:6 bitplane driving scheme provided by Holoeye support team to minimize flicker often seen at higher driving schemes [5]. The source for the interferometric and diffraction based methods is a 532 nm solid state laser (Coherent Verdi 6) which is spatially filtered using a pinhole and collimated. A ThorLabs 530 nm Light Emitting Diode (LED) is used for the cross polarizing diagnostic to eliminate undesirable interference phenomena that arise from coherent illumination.

Interferometry

In this study we use a Michelson interferometer as shown in Figure 2. The SLM takes the place of the test mirror and a reference mirror with a flatness of $\lambda/20$ or better is mounted on a nano-positioning stage with closed loop feedback control (ThorLabs NFL5DP20S). A reduction telescope with magnification 0.66X images the SLM onto the CMOS camera.

Of the variety of shifting methods available, we chose the 5-step, Hariharan, method as a suitable compromise between measurement time and accuracy [6]. The nano-positioning stage was moved in $\delta = \pi/2$ increments, collecting 5 intensity measurements from the CMOS camera.

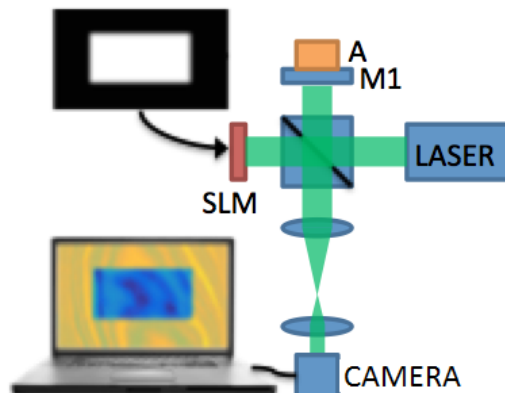


FIGURE 2. Michelson Interferometer with the SLM as the device under test. Reference mirror, M1, is on a nano-positioning actuator, A. A rectangle with variable 8-bit values is displayed on the SLM with a constant 0 grey value perimeter. An interferogram is captured using the camera and is converted into a phase map after using a 5-step phase shifting technique.

To measure the phase change for each grey value, a rectangle with constant grey values was displayed on the SLM, surrounded by a constant 0 grey value. The phase, ϕ , at each grey value was then calculated using

$$\phi = \tan^{-1} \left(\frac{2(I_2 - I_4)}{2I_3 - I_5 - I_1} \right) \quad (1)$$

where I_1 , I_2 , I_3 , I_4 and I_5 are image of intensity taken at $\delta = 0, \pi/2, \pi, 3\pi/2$, and 2π , respectively.

To recover the phase measurement at each gray value, first, any high-frequency image content was removed by Fourier low-pass filtering. Then a MATLAB routine was used to unwrap the phase profile and remove any tilt of the SLM. The phase offset was removed by subtracting the average phase on the outer perimeter of the rectangle. The resulting average phase value within the rectangle thereby yields the actual phase change, $\Delta\phi$, for each grey value.

One complication for determining the phase change between the zero valued perimeter and the inner rectangle is that the results must be unwrapped from the $-\pi$ to π domain. Figure 3 shows the phase data with grey value 172 taken using the 5-step method (top) compared to the phase map after unwrapping using the MATLAB built in unwrap function, removing tilt, and subtracting the phase profile for a gray value

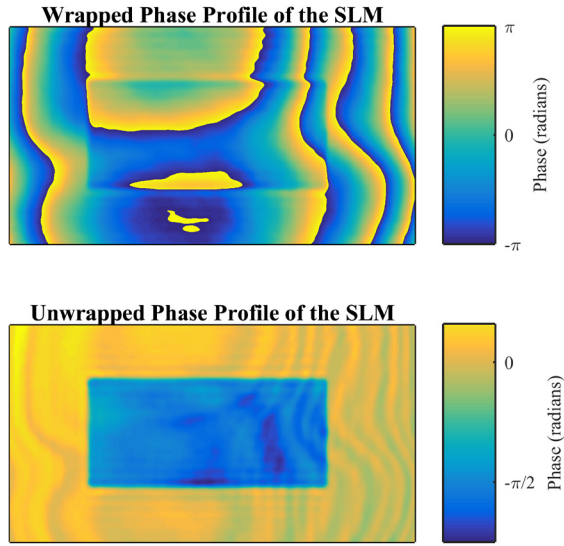


FIGURE 3. Wrapped (top) and unwrapped (bottom) phase of the SLM with 172 grey value rectangle in the center and 0 grey value outside. The unwrapped phase was calculated by unwrapping the phase discontinuities and removing the tilt, and subtracting phase variance due to the non-flat surface of the SLM.

of 0 (bottom). The standard MATLAB unwrapping algorithm has trouble handling measured phase values around π and 2π , so this aspect of the analysis will need to be made more robust in the future.

Figure 4 shows the phase change as a function of grey value displayed on the SLM. The phase is displayed between $-\pi$ and π and values around π and 2π needed to be determined manually. Our measurement implies that the experimental values for phase, as measured by the Michelson Interferometer, do not match with the encoded calibration table.

Polarization Approach

In a similar configuration, we implemented two linear polarizers to measure the relationship between phase and grey value as in [7, 8]. Figure 5 shows the experimental apparatus. The first linear polarizer, LP_1 is oriented at 45° to the horizontal. The SLM delays the phase of only one polarization, the one oriented along the horizontal; therefore the phase reflected from the SLM is elliptical. The second polarizer, LP_2 is oriented at $\pm 45^\circ$ to the horizontal to correct for the light that was unchanged by the SLM. The phase change, $\Delta\phi$, induced by the liquid crystals

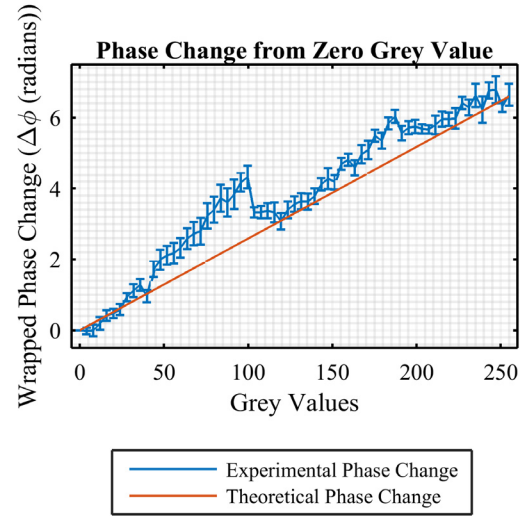


FIGURE 4. Change of phase relative to zero grey value displayed on the SLM. The calibrated gamma table we used was linear 2.1π phase range.

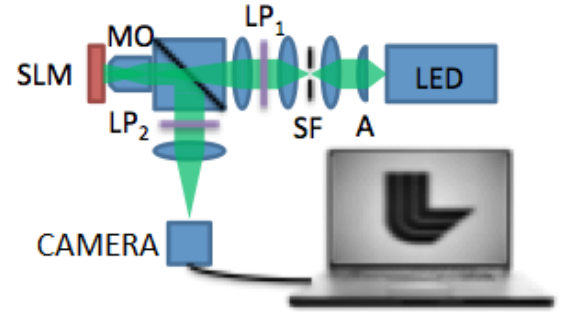


FIGURE 5. Phase measurement setup using linear polarizers. The first linear polarizer P_1 is oriented at 45° . The SLM delays the horizontally polarized light, the light is reflected off the back plane and is delayed again by the SLM. The light is reflected off the beam splitter and travels through a second polarizer, P_2 oriented at $\pm 45^\circ$. The image of the resulting intensity is displayed on the CMOS camera.

can be calculated using Jones matrices from the intensity values of the recorded image. The Jones Matrix of a reflective SLM, derived from [9], is

$$\begin{pmatrix} e^{-i2\phi} & 0 \\ 0 & 1 \end{pmatrix} e^{-i2kn_0d} \quad (2)$$

where d is the thickness of the liquid crystal layer and n_0 is the refractive index of the SLM with no induced phase delay.

Following through the system using Jones matrices and an incoherent light source with electric field, E ,

$$E = \begin{pmatrix} E_x e^{i\phi_x} \\ E_y e^{i\phi_y} \end{pmatrix} \quad (3)$$

where E_x , E_y , ϕ_x and, ϕ_y are the independent amplitudes and complex phases, respectively, for the components of light perpendicular to the propagation direction, the resulting electric field, E_{system} , is

$$E_{system} = \frac{1}{4}(e^{i\phi} \pm 1) \begin{pmatrix} E_x e^{i\phi_x} + E_y e^{i\phi_y} \\ \pm E_x e^{i\phi_x} \pm E_y e^{i\phi_y} \end{pmatrix} \quad (4)$$

where the positive equation arises from polarizers rotated parallel to each other and the negative equation is a result of perpendicular polarizers. The effect is a phase shift of π which is addressed when the intensity is simplified below. The intensity for each pixel on the SLM is given by

$$I(\Delta\phi) = E_{system}^\dagger E_{system} \quad (5)$$

$$= \frac{1}{8}(E_x e^{i\phi_x} + E_y e^{i\phi_y})^2 (e^{i\Delta\phi} \pm 1)^2 \quad (6)$$

which can be reduced to

$$I(\phi) = I_0 \cos^2\left(\frac{\phi}{2}\right) \text{ or} \quad (7)$$

$$I(\phi) = I_0 \sin^2\left(\frac{\phi}{2}\right) \quad (8)$$

where I_0 is a constant with change of ϕ and only depends on incident amplitude and phase. This result shows the phase response of each pixel when imaged through the linear polarizers can be calculated by normalizing the intensity and solving for ϕ .

This approach can be easily adapted to measuring the spatially-varying phase response over the area of the SLM. Accordingly, as a first step, we measured the phase of one portion of the SLM at high magnification using a Nikon 40X Plan Fluor microscope objective and a 75 mm tube lens resulting in a field of view that images 104x84 pixels on the SLM. The use of high NA optics allows recording of sub-pixel information. Similarly to the interferometric approach, a 100x80 pixel rectangle at varying gray values was displayed in the center of the field of view, and the measured intensity of the resulting image was extracted, using a custom MATLAB routine. Figure 6 shows the measured intensities as a function of gray value for a

single SLM pixel, compared with the analytically-predicted functional form.

These measurements match well to the analytical expression, in which the intensity of a modulated phase through 45° linear polarizers is proportional to $\cos^2(\phi/2 + \delta)$. Work is ongoing to map the full phase response of the SLM area on a pixel-by-pixel basis.

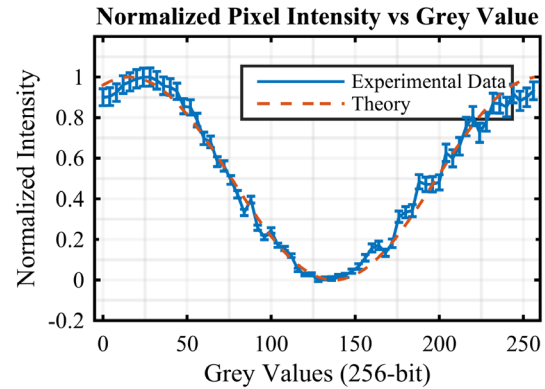


FIGURE 6. Experimentally validated normalized intensity versus grey values for a single pixel. The theoretical intensity is proportional to $\cos^2(\phi/2 + \delta)$ where δ is a phase shift.

Diffraction

Several authors display a binary grating on an SLM because the phase delay from addressed values can be analytically extracted from the intensity of the diffracted orders using the Fourier transform [4, 10, 11]. When a perfect binary phase grating is displayed on the SLM and is passed through a Fourier lens, the normalized power of the zeroth and first orders are proportional to

$$P_0(\phi) = \frac{1 + \cos(\phi)}{2} \text{ and} \quad (9)$$

$$P_{\pm 1} = \left(\frac{2}{\pi}\right)^2 \frac{1 - \cos(\phi)}{2} \quad (10)$$

respectively, where P_0 and $P_{\pm 1}$ are the normalized power and ϕ is the phase modulation from the SLM [10]. Ronzitti suggests that the output phase from the SLM has inter-pixel phase effects and results in a sinusoidal rather than crenel-shaped phase response. With that modification, the resulting power for the zeroth and first diffraction orders has the forms

$$P'_0(\phi) = \left(J_0\left(\frac{\phi}{2}\right)\right)^2 \text{ and} \quad (11)$$

$$P'_1(\phi) = \left(J_1\left(\frac{\phi}{2}\right)\right)^2 \quad (12)$$

respectively, where J_0 and J_1 are the zero and first order Bessel functions, respectively.

Figure 7 shows the setup used to measure the intensity of the diffraction orders from the SLM when a linear phase grating is applied. A binary grating with a period of two pixels was displayed on the SLM. A Fourier lens with focal length 250 mm was used one focal length from the SLM. The power at the zeroth and first order were taken at various grey values using a Thorlabs digital power meter (PM121D). Figure 8 shows the normalized power measurements for the zeroth and first diffraction orders. Theoretical values for a crenel-shaped and sinusoidal response are shown for comparison.

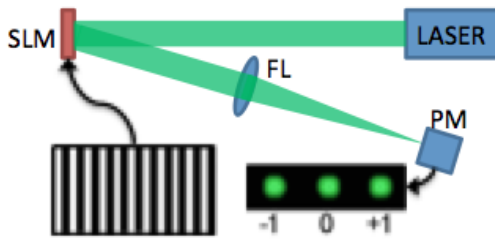


FIGURE 7. Setup to measure the intensity of the diffractive modes from the SLM. A linear diffraction grating is displayed on the SLM. A Fourier lens, FL, is placed one focal length away from the SLM. Each mode is measured using the power meter, PM.

The phase measured using the diffraction method corresponds to the overall phase change of the entire illuminated area of the diffraction grating (in this case the whole SLM), including any inherent physical defects. Preliminary results suggest that more refined calibration and analytical models are necessary due to physical defects in the SLM and approximation of phase response shape. Engström suggests splitting the SLM into regions to measure the phase delay at each particular region of the SLM [4]. Yu suggests that the response of the SLM should account for the distance between the pixels [11]. Further analysis will be carried out to approximate the phase response using images taken with the linear polarizer setup mentioned above.

RESULTS AND DISCUSSION

Figure 9 compares the phase response measured from each method with their associated uncertainties. The phase for the polarizer method was calculated using equation 7 and solving for ϕ . Phase for the zeroth and first order diffraction

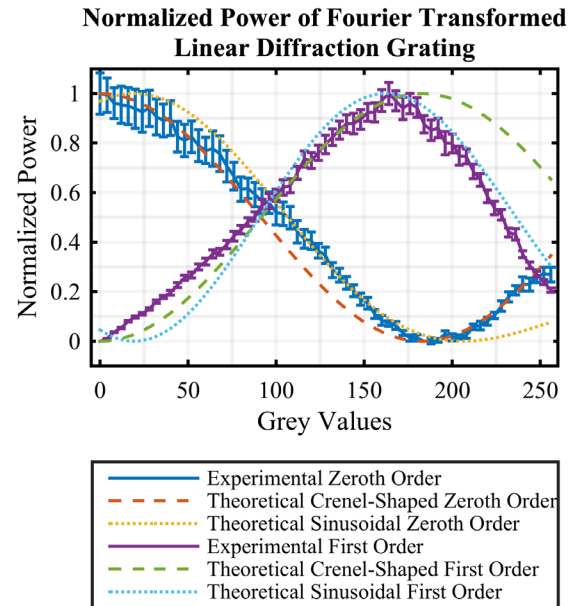


FIGURE 8. Measurements of normalized power versus grey values of the zeroth and first diffraction order from a linear grating. The theoretical values for a crenel-shaped and sinusoidal response are shown with fitting parameters for phase offset and are normalized.

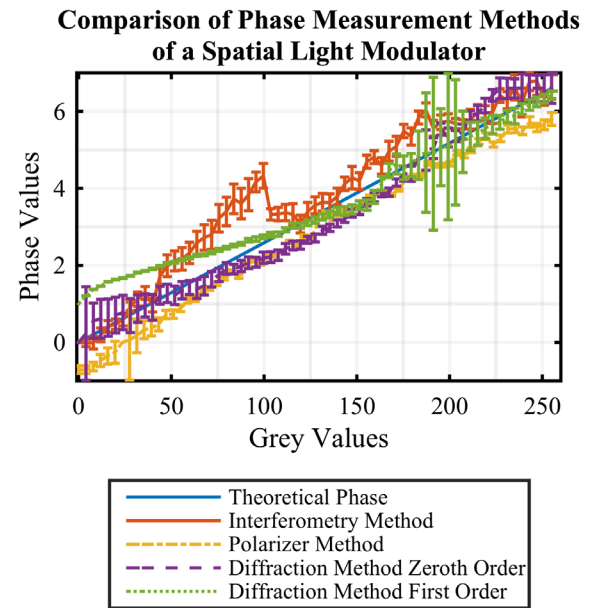


FIGURE 9. Graph comparing phase measured by each method

TABLE 1. Method Comparison

Parameter	Phase Shifting Interferometry	Linear Polarizer	Diffraction
Measurement Type (over SLM face)	Continuous	Discrete	Overall
Light Source	coherent	incoherent	coherent
Angle of Incidence	0	any	any
Advantages	can measure flatness of SLM, can calibrate each pixel	can calibrate each pixel, one measurement per grey value, absolute measurement, low spacial footprint	can be measured in Fourier domain, can create subsection of SLM and calculate phase delay of region
Disadvantages	No discretization of pixels, must unwrap phase, only relative measurement compared to zero value, low confidence at π and 2π phase values	Highly data intensive, must have at least π range to extract phase, must have incoherent source	measures overall phase, cannot resolve single pixel phase variation

was calculated numerically using Equations 11 and 12, respectively, and solving for ϕ . The uncertainties for the polarization and diffraction method were calculated by taking the derivative of equations associated with each method and solving for $d\phi$.

We have demonstrated three concepts to measure the fundamental errors affecting the phase in a spatial light modulator. Table 1 some considerations for comparing the performance of interferometry, polarization, and diffraction methods to measure the phase variance on a spatial light modulator. Ongoing work is being performed to rigorously quantify and systematically correct for errors associated with non-flat back plane and nonlinear electrically induced birefringence. With this information we will be able to construct an optimal *in situ* calibration apparatus for our system to use with optical tweezers and holographic lithography.

ACKNOWLEDGMENTS

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UNCERTAINTY QUANTIFICATION OF AN X-RAY SYSTEM VIA A 0D MODEL USING A SYSTEMS APPROACH

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INTRODUCTION

Dimensional metrology via X-ray computed tomography (XCT) is yet limited by the higher requirements and standards of spatial/structural resolution and accuracy compared to those required in traditional applications such as medical and material characterization [1]–[3]. XCT measurements are influenced by various parameters resulting in additional error sources, which are not present in conventional techniques including standardized tactile methods (e.g. coordinate measuring machines, CMMs). Those uncertainties are intrinsic of the experimental setup and the tomographic reconstruction algorithms implemented for data post-processing. Some of these error sources in the environment, x-ray source, detector and other components include the temperature, humidity, mechanical vibrations, beam hardening effects, quantum efficiency, threshold determination, and others [1], [2]. The quantification of uncertainty on X-ray measurements has been implemented in previous work using multiple calibration samples and XCT systems [4]–[8] in order to improve the capabilities of XCT as a dimensional metrology tool. Other studies have focused on characterizing error sources of specific

components from the X-ray system to optimize the measurement [9]–[13].

A new uncertainty model framework that attempts to characterize and quantify the uncertainty from the dominant factors in each data acquisition hardware and software component is presented with the goal of establishing and optimizing XCT systems for metrology applications. The ongoing modeling effort focuses on uncertainties in the object transmission value measured at a single pixel, which is a 0D point at the detector. The same framework will next be applied via 1D and 2D models with the goal of capturing all of the uncertainties in radiographs. This 0D uncertainty model includes the noise and sensitivities from mechanical vibrations, temperature variations, the electrical signal, and count rate at the source as well as at the detector. All of these uncertainties could potentially affect the performance of an X-ray system at a specific wavelength (energy level) for a single pixel measurement. The model also considers data acquisition and calibration factors such as detector pixel cross talk, pixel binning and detector background noise. In this study, the results of a simplified uncertainty model are compared to intrinsic statistical fluctuations

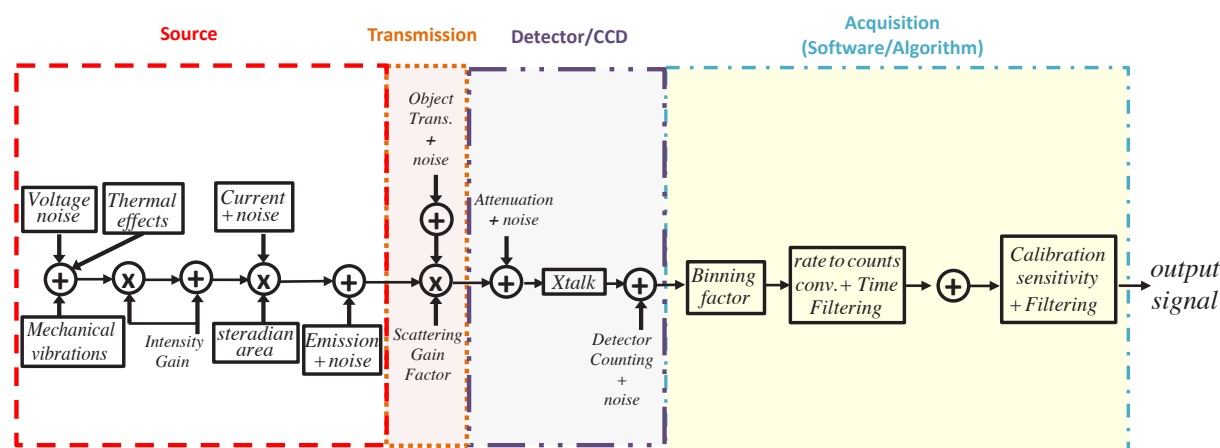


Figure 1: Signal variation flow graph representation with all parameters

including Poisson and binomial noise which occur during X-ray emission and object transmission. Thus, this study validates the 0D uncertainty model by comparing the results with conventional counting statistics (i.e. the framework of statistical analysis required to process the results of nuclear counting experiments) and provides a detailed explanation of the approach to calculate the uncertainty components.

BACKGROUND

The model developed herein takes into account the contributions to signal variation as measured at a single pixel, including uncertainties and sensitivities from different sources and for each data acquisition X-ray component applying a systems approach [14]. The model is essentially a 0D since the uncertainty of the transmission for each pixel at a given projection (i.e. radiograph) is calculated. The 0D model considers only errors which have no larger scale form, and thus do not require higher dimensional modeling. Such higher dimensional errors will be captured in later models. Inasmuch, the object is assumed to be a block of uniform thickness from the perspective of the studied pixel 'point' on the detector.

Figure 1 displays the overall parameters included in the model for each X-ray system component in the form of a signal variation flow graph. For instance, the source includes mechanical vibrations at the source and with respect to the detector, ambient and component temperature effects, electrical current noise and source emission variation which could affect the final emission of X-rays before they are recorded by the detector. Similarly, other noises are acquired in the transmission and detector regime due to the object attenuation and

scattering as well as the efficiency and count noise of the detector. The signal variation flow graph also captures effects due to detector pixel cross talk, dark field offset (i.e. background noise of the detector) and a binning parameter in addition to a calibration term that normalizes the final response of the model output signal (Ψ_m).

Furthermore, the signal variation flow graph depicts the use of low and high pass filtering to create a bandwidth of the noise parameters in the frequency domain when characterizing the power spectral density functions.

APPROACH

A simplified 0D model that includes only the emission and object transmission uncertainty is evaluated and utilized to validate the Poisson and binomial noise inherent of nuclear counting statistics. The model is validated by three cases comparing (i) the emission noise assuming no object to the Poisson noise, (ii) the object transmission noise to the binomial noise and (iii) the total uncertainty of the model to the Poisson noise of the attenuated counts. This model is representative of an X-ray system which consists of a point X-ray source and semiconductor detector, depicted in Figure 2a. The mechanical, electrical, and thermal noise are assumed to be negligible and non-existence in the case that the system is fixed and that the source is an encapsulated radioactive material. The scattering factor is ignored since the source-to-detector distance is short enough and any noise related to the detector (i.e. σ_{η_l} , σ_{ndc} and Δ_{ndc}) is insignificant assuming the efficiency is sufficiently high. In addition, the cross-talk and binning factor are also not included due to the simplified X-ray system setup. The final simplified 0D model is displayed

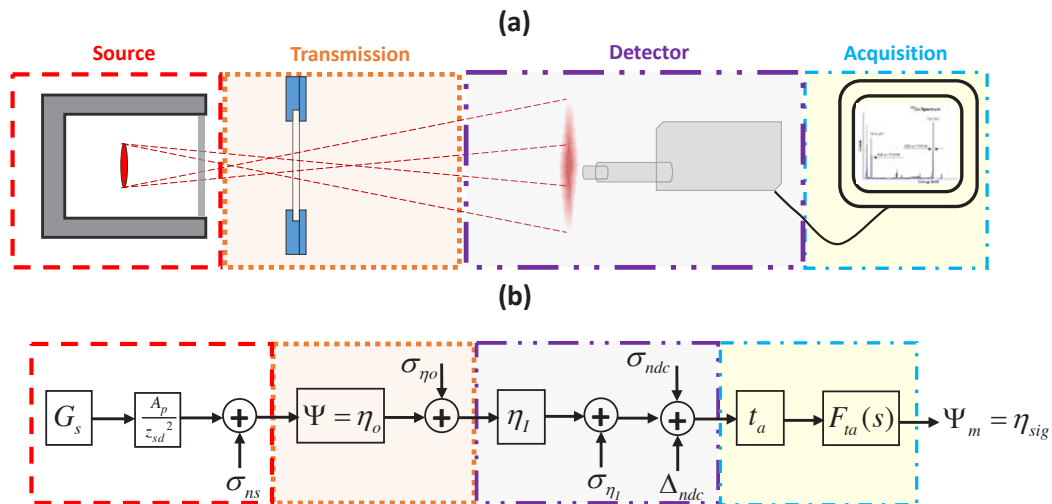


Figure 2: (a) X-ray gage (single pixel) system schematic and (b) 0D uncertainty model block diagram representation of (a)

in Figure 2b and the final output signal in unit of counts can be written as,

$$\psi_m = \left(\left(\sigma_{ns} + \frac{A_p}{z_{sd}^2} G_S \right) \eta_o + \sigma_{\eta_o} \right) t_a F_{ta} \quad (1)$$

, where A_p is the pixel area (active area) as a function of the source coordinates, z_{sd} is the distance from source to the detector, G_S is the X-ray intensity (flux) gain factor term, t_a is the exposure time (i.e. integration time), F_{ta} is the low pass filter, σ_{ns} and σ_{η_o} are the noises from the emission and object transmission, respectively. The intensity gain factor G_S can be calculated from (1) by setting η_o equal to 1 (i.e. there is no object as all photons from the emission pass through) and ψ_m equal to the number of counts measured (plotted in in Figure 3). Assuming a small time interval and neglecting all noises G_S is:

$$G_S = \frac{n_{cts} \cdot z_{sd}^2}{A_p t_a} \quad (2)$$

, where n_{cts} is the number of counts (Figure 3).

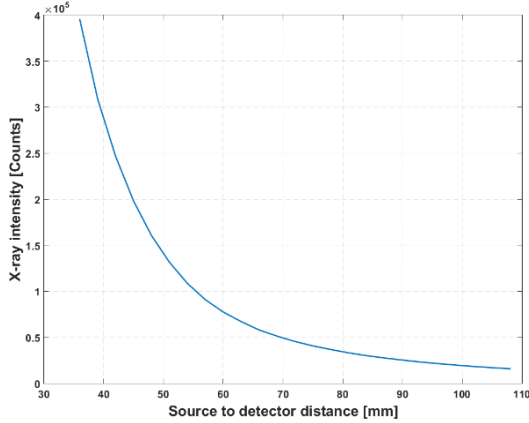


Figure 3: Number of counts as a function of source-to-detector distance

F_{ta} is a low-pass filter mimicking the frequency attenuation due to an integration over time t_a . This is formulated as a function of s , which is equal to $2\pi if$ using the standard Laplace transform terminology, resulting in:

$$F_{ta}(s) = \frac{1 - e^{-st_a}}{s \cdot t_a} \quad (3)$$

The spectral density of each noise is scaled by their respective frequency dependent sensitivities and geometrically summed to obtain the full system noise spectral density:

$$S_{\psi_m}(f) = \sum_n \left| \frac{\partial \psi_m(2\pi if)}{\partial \sigma_n} \right|^2 S_n(f) \quad (4)$$

An estimate of the noise variance is calculated by integrating the system spectral noise density over a frequency range. Then, the total uncertainty is expressed as

$$\sigma_{\psi_m} = \sqrt{\int_{f_{\min}}^{f_{\max}} S_{\psi_m}(f) df} \quad (5)$$

RESULTS

The results in this study quantify the emission and object transmission noise components of the model, shown in Figure 2b as σ_{ns} and σ_{η_o} .

The calculated values are then compared to the Poisson and binomial noises which are conventionally utilized in nuclear counting statistics for the emission and object transmission, respectively. The spectral density functions of the emission and the transmission can be formulated as:

$$S_{ns}(f) = 2 \frac{G_S A_p}{z_{sd}^2} \quad (6)$$

$$S_{\eta_o}(f) = 2 \frac{G_S A_p}{z_{sd}^2} \eta_o (1 - \eta_o) \quad (7)$$

The emission uncertainty of the output signal using (5) and (6) is given by:

$$\begin{aligned} \sigma_{\psi_{ns}} &= \sqrt{\left(\int_0^\infty |F_{ta}(f)|^2 df \right) |\eta_o t_a|^2 2 \frac{G_S A_p}{z_{sd}^2}} \\ &= \eta_o \sqrt{\frac{G_S A_p}{z_{sd}^2} t_a} \end{aligned} \quad (8)$$

Figure 4 displays the results from (8), when $\eta_o = 1$ (i.e. no object is present), as a function of source-to-detector distance. As it can be seen, the results from the OD model are identical to those of the conventional noise quantification for

the emission. This can also be verified since the Poisson noise from the measurement is:

$$\sigma_{Poisson} = \sqrt{n_{cts}} = \sqrt{\frac{G_s A_p}{z_{sd}^2} t_a} \quad (9)$$

It is important to note that the frequency range utilized was over the entire domain. Thus, the only term that is a function of frequency and integrated over the domain is:

$$\int_0^\infty |F_{ta}(f)|^2 df = \frac{1}{2t_a} \quad (10)$$

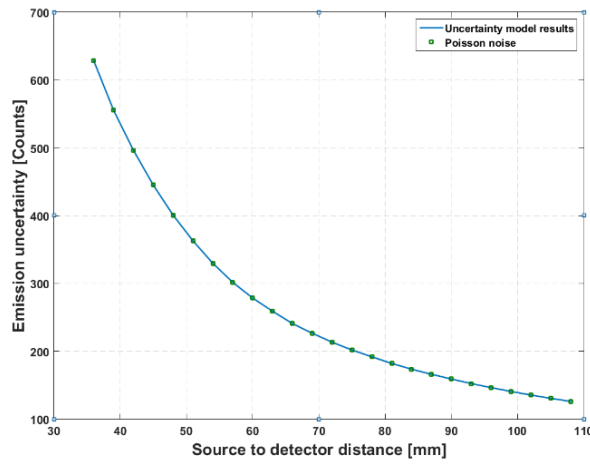


Figure 4: Uncertainty model results compared to Poisson noise from the X-ray emission as a function of the source-to-detector distance

Similarly, the uncertainty of the object transmission is calculated from (5) and (7) and can be simplified to:

$$\begin{aligned} \sigma_{\psi\eta_o} &= \sqrt{\left(\int_0^\infty |F_{ta}(f)|^2 df \right) |t_a|^2 2 \frac{G_s A_p}{z_{sd}^2} \eta_o (1 - \eta_o)} \\ &= \sqrt{\frac{G_s A_p}{z_{sd}^2} \eta_o (1 - \eta_o) t_a} \end{aligned} \quad (11)$$

Figure 5 depicts the results from the model for the object transmission and its corresponding binomial noise at different source-to-detector distances. The results show that the model captures accurately the binomial noise, which is mostly used in counting statistics when there is a higher chance for a random event to occur and is usually defined as:

$$\begin{aligned} \sigma_{Binomial} &= \sqrt{n_{cts} \eta_o (1 - \eta_o)} \\ &= \sqrt{\frac{G_s A_p}{z_{sd}^2} \eta_o (1 - \eta_o) t_a} \end{aligned} \quad (12)$$

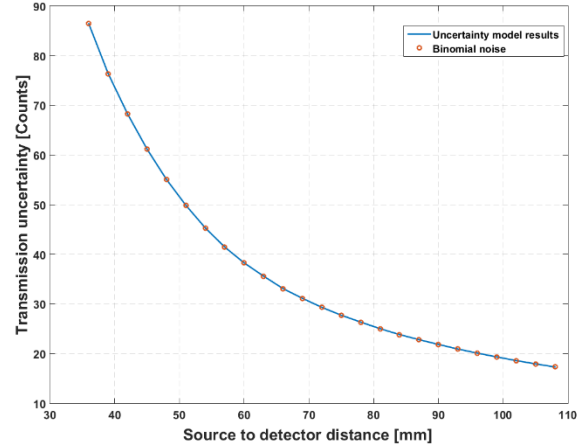


Figure 5: Uncertainty model results compared to binomial noise from the object transmission as a function of the source-to-detector distance

The last case consisted of comparing the combined uncertainty of the simplified model with the Poisson noise of the attenuated counts. This can be calculated using (4) - (7) in order to add all uncertainty components. This results in:

$$\begin{aligned} \sigma_{\psi m} &= \sqrt{\sigma_{\psi ns}^2 + \sigma_{\psi \eta_o}^2} = \sqrt{\eta_o^2 \left(\frac{G_s A_p}{z_{sd}^2} \right) + \eta_o (1 - \eta_o) \left(\frac{G_s A_p}{z_{sd}^2} \right)} \\ &= \sqrt{\left(\frac{G_s A_p}{z_{sd}^2} \right) \eta_o} = \sqrt{n_{cts} \eta_o} \end{aligned} \quad (13)$$

The total uncertainty from the model in (13) is essentially the Poisson noise of the attenuated counts which demonstrates that the model captures precisely the conventional methods used to measure the uncertainty in X-ray radiograph measurements.

CONCLUSION

The model is compared to theoretical noise propagation for counting statistics and shown to be able to accurately represent the propagation of uncertainty through the X-ray system. Although a simplified version of the model was presented, this method is designed to be capable of extension to cover higher

dimensional uncertainties with a target of mapping the full range of variation observed in X-ray radiographs.

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Virtual Instrument Connecting Global-Local Measurements

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BACKGROUND

Tolerance callouts for feature, such as hole locations, profiles, etc., are often widely distributed on large parts and assemblies. While an instrument may be able to measure the detail, the relationship of measurement within the tolerancing scheme might not be determinable by that instrument because its field-of-view is too small. A second “global” instrument may be required for determining the location and orientation of the “local” instrument within the part’s coordinate frame.

The typical method for combining the local and global systems uses targets associated with the local system which the global system measures. These targets can be a constellation of targets about the local system [1], or specialize targets such as an Automated Precision Target [2] or a Leica T-Mac [3]. Figure 1 shows magnetic mounts for a constellation of spherically mounted retro-reflectors (SMRs) about a local measurement system. Figure 2 shows one of two T-Macs monitoring the head of an NC machine. The challenge with most targeting schemes is their limited acceptance angle. In figure 1 the SMR will need to be rotated somehow, which interferes with automation. T-Macs used on the NC machine shown in figure 2 limit the rotation of the head. This paper concentrates on issues related to using SMRs, realizing that concepts should apply to other local targeting schemes.

In a simplistic fashion the radial measurement uncertainty of the local and global systems could be combined via root-sum-square (RSS).

$$U = \sqrt{U_L^2 + \frac{U_G^2}{\sqrt{n}} + \frac{U_P^2}{\sqrt{m}}} \quad [\text{eq 1}]$$

where: U_L = the uncertainty for the local measurement system,

U_G = the uncertainty for the global measurement system at the constellation,

n = the number of targets in the constellation,

U_P = the uncertainty for the global measurement system at the part coordinate frame,

m = the number of target in the part reference frame.



FIGURE 1. Magnetic Holders for Constellation and Reference SMRs [1]

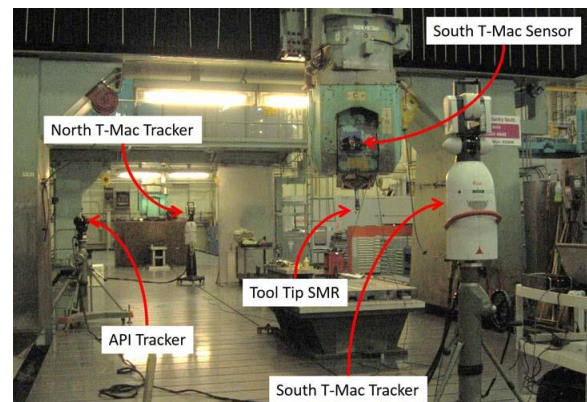


FIGURE 2. Dual T-Mac Measurement of NC Machine [1]

Missing from equation 1 is the variation in the orientation of the constellation which is another aspect of the uncertainty when fitting measured values to the constellation of targets. It creates uncertainty in the separation between the constellation centroid – the locus of transformation rotation and translation – and the local measurement system origin. And more importantly, the pointing of the local measurement system toward the measured object shifts with this angular variation. Had a gimbal been used to effect this angular pointing variation, its uncertainty would be obviously and easily included in equation 1. The constellation of targets partially obscures this effect, partially because the uncertainty is not a simple calculation – it is a function of the uncertainty of the constellation measurements, number and configuration of constellation targets measured, and distance to the constellation centroid to the measured object. This interface between the global and local instruments can be thought of as a virtual instrument (VI) because it is akin to pointing a gimbal and creates an additional term in equation 1. Figure 3 conceptualizes the various coordinate frame relationships – the part local, global and virtual instruments. Examining the parameters of the VI can help drive both equipment design and measurement processes for minimizing the total measurement uncertainty.

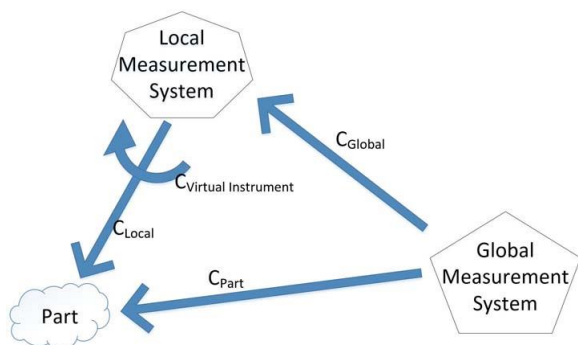


FIGURE 3. Measurement Coordinate Frames

CONSTELLATION OF POINTS – MONTE CARLO

A good way to study the VI uses Monte Carlo methods. Various scenarios can be applied to the interface – number and location of constellation targets as well as the locations and orientation of the local system within that constellation. And finally, it is likely that not all of the constellation will be measured, so the impact

of having a constellation subset can be modeled. For simplicity, this VI study omits the uncertainty related to global measurement of the part's coordinate frame.

Figure 4 shows a constellation and location of the local system. Eight points at the corners of the cuboid form the constellation of targets. The local system's origin and primary line-of-sight (LOS) are also shown. The local system primary LOS is a vector whose length is typical measurement range for the local system. The base of the LOS vector predicts the position variation for the local instrument origin. The tip of the LOS vector, often called an indirect or inferred measurement point, predicts the pointing variation of the local measurement on the measured object, which is the missing VI in equation 1.

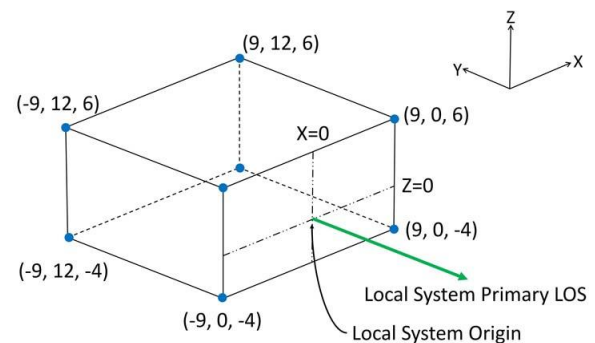


FIGURE 4. Local Measurement System and Constellation of Targets

In figure 4, the local system origin is not at the centroid of the cuboid – generally there is consideration for clearance between the measurement system, target constellation and measured object. Because the local system origin is offset from the cuboid centroid, there will be positional variation of the origin for the local measurement system. Even if the local origin were at the cuboid centroid, the likelihood of measuring the entire constellation of points would be doubtful, so the local origin will generally vary from the actual centroid of constellation measurements.

The Monte Carlo analysis uses a coordinate transformation calculation of the rigid body model of the constellation and local LOS vector (table 1, columns 1 – 3). The constellation centroid and its unit vector in the direction of the local instrument LOS are included in the transformation computation, table 1, lines 9 and

10. The local instrument origin and its primary LOS are the 11th the 12th lines in table 1. Nominal values from the constellation are created from cuboid values. From these, simulated measurements are constructed by applying a random Gaussian envelope to the uncertainty of the global measuring instruments (table 1 columns 4 – 6). Weighing, either uniform of 1 or based on the global system variance, is shown in the 7th column of table 1. The resulting least squares transformed values (table 1, columns 8 – 10) for the local origin and LOS vector are used to evaluate the virtual instrument connecting the global and local systems. Note: In this evaluation the order of reference (constellation) and measured values have been swapped. Ordinarily the measured values are transformed into the reference values. In this case, the variation in the LOS vector is of interest, so the order is swapped in order to extract the motion of the LOS vector. The analysis was performed using MathCAD for generating the random variables as well as tracking the Monte Carlo trial results. The least squares coordinate transform [4, 5, 6] was implemented as a user defined Excel function.

TABLE 1. Typical Monte Carlo ith Trial.

Constellation			Measured			Weight	Transformed		
X	Y	Z	X	Y	Z		X	Y	Z
9	0	-4	9.0009	0.0013	-3.9987	1	8.9999	0.0004	-3.9989
-9	0	-4	-9.0042	-0.0012	-3.9990	1	-9.0001	-0.0001	-3.9995
9	0	6	9.0014	0.0000	6.0018	1	8.9996	0.0010	6.0011
-9	0	6	-9.0001	0.0018	5.9993	1	-9.0004	0.0005	6.0005
9	12	-4	9.0004	12.0018	-3.9985	1	8.9996	12.0004	-3.9997
-9	12	-4	-9.0002	11.9990	-3.9996	1	-9.0004	11.9999	-4.0003
9	12	6	9.0007	12.0003	5.9993	1	8.9993	12.0010	6.0003
-9	12	6	-9.0022	12.0007	5.9985	1	-9.0007	12.0005	5.9997
0	6	1					-0.0004	6.0005	1.0004
0	5	1					-0.0004	5.0005	1.0004
0	0	0					-0.0002	0.0004	0.0008
0	-16	0					0.0002	-15.9996	0.0018

CONSTELLATION OF POINTS -- ANALYSIS

Global measurement uncertainty for the constellation translates into uncertainty in both the location and the orientation of the constellation. Assume combining the measurement standard uncertainties (i.e. 2-sigma) of an area array scanner (e.g. $U_L = 0.001$ in) with that of a tracking interferometer (e.g. $U_G = 0.0005$ in + 10ppm × range). Measuring the figure 1 constellation of targets at a range of 25ft the global system will have a standard uncertainty of ±0.0035 inch. Resulting Monte Carlo analysis indicates a 2-sigma variation at the constellation centroid of about ±0.0013 inch in X, Y and Z (see table 2). The variation of the local measurement system origin is slightly larger because there is angular variation in the separation between the constellation centroid

and local system origin. The average angular variation of the constellation is about ±54 μrad and ±80 μrad in the X and Z directions, respectively – the X length being longer than the Z length, so is more “accurate”. Assuming a local system measurement range of 16 inches results in the tip of the LOS vector having a variation of ±0.0027, ±0.0012, and ±0.0038 inch, in X, Y and Z, respectively; the average is 0.0026 inch.

The average 2-sigma of the centroid position for any X, Y or Z term is approximately 0.0013 inch. This roughly corresponds the radial standard deviation of the mean for the constellation measurement – ±0.0035/sqrt(8) = ±0.0012 inch – which is used in equation 1. The variation in the LOS vector is the missing element when merely combining the variation of the global and local measurement systems: so a better estimate for the combined measurement uncertainty is:

$$U = \sqrt{U_L^2 + \frac{U_G^2}{\sqrt{n}} + \frac{U_P^2}{\sqrt{m}} + VI(range)^2} \quad [\text{eq 2}]$$

where: $VI(range)$ is the uncertainty associated with the LOS vector – the virtual interface.

TABLE 2. Typical Monte Carlo 2-Sigma Variation for Figure 1 Constellation at 25 ft.

	Centroid	Local Origin	Angle (μrad)	VI
X	0.0013	0.0014	54	0.0027
Y	0.0012	0.0012		0.0012
Z	0.0013	0.0016	80	0.0038
Avg	0.0013		Avg	0.0026

Changing the size of constellation impacts the pointing angle but not the variation of the constellation centroid. For example, if the size of the constellation were doubled the variation in the centroid will remain the same, but the angular variation will be halved (see tables 2 and 3). This size change can impact the geometry – the distance from the constellation centroid to the local origin may be longer and thus have increased variation. Likewise, the distance from the constellation centroid to the tip of the LOS vector may be longer, counteracting the increased angular accuracy. The result is generally a reduced angular uncertainty, but not necessarily an inverse relationship to the increase in the constellation size.

TABLE 3. Typical Monte Carlo 2-Sigma Variation for Figure 1 Constellation Doubled in Size at 25 ft.

	Centroid	Local Origin	Angle (μ rad)	VI
X	0.0014	0.0016	26	0.0022
Y	0.0014	0.0014		0.0014
Z	0.0012	0.0016	39	0.0026
Avg	0.0013		Avg	0.0021

A practical issue with a constellation of targets is that they are typically not all measured, either because some targets are occluded or because of process time issues – omit targets and save flow time! If the six constellation targets at the end of the cuboid nearest the global measurement system are measured then the measurement centroid of the transform is shifted, its positional uncertainty increases, and the angular uncertainty increases. The result is that the uncertainty at the LOS vector is increased. Table 4 shows a typical least squares computation of six targets. Compared to table 3, table 4 values are larger.

TABLE 4. Typical Monte Carlo 2-Sigma Variation for 6-Target Constellation at 25 ft.

	Centroid	Local Origin	Angle (μ rad)	VI
X	0.0014	0.0015	66	0.0030
Y	0.0014	0.0015		0.0015
Z	0.0014	0.0016	106	0.0043
Avg	0.0014		Avg	0.0029

Graphs of uniformly weighted and variance weighted uncertainties for six target constellations are shown in figures 5 and 6, respectively. Obvious is the global and VI measurement uncertainties increasing with range. And with a single global measurement system, the weighting scheme is of little consequence, as both results are similar. It is clear from figures 5 and 6 that the uncertainty contribution from the VI is a major contributor and must be accounted for in the total measurement uncertainty.

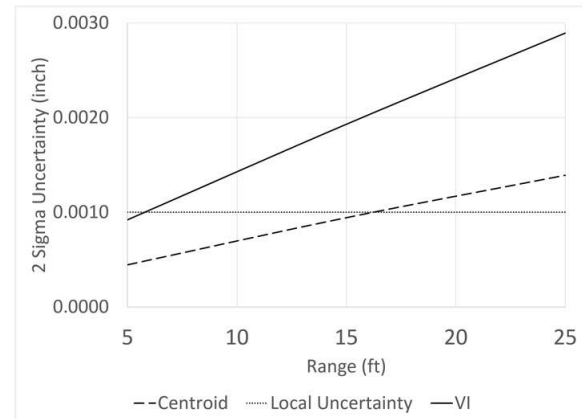


FIGURE 5. 6-Target Constellation 2-Sigma Uncertainties with Uniform Weighting (=1) for the Centroid, Local Origin and Virtual Instrument.

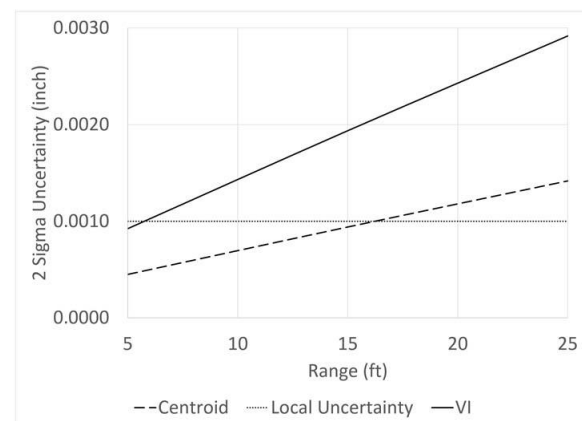


FIGURE 6. 6-Target Constellation 2-Sigma Uncertainties with Variance Based Weighting for the Centroid, Local Origin and Virtual Instrument.

The single global measurement system exemplified above has uncertainties as a functions of range. Likewise, the VI is also a function of range. A way to mitigate these functions of range is to use multiple global measurement systems. There are many ways to configure multiple global measurement systems. The following assumes the two systems are separated by 50 ft with the object between (figure 7). Each of the global systems measures 6 constellation targets at its end of the constellation cuboid. Uniform and variance based weightings are applied when fitting the two sets of measurements. Tables 5, 6 and 7 show resultant Monte Carlo results for two different positions of the object between the two global measurement systems. Table 5 is for the

part centered between the global measurement systems, while tables 6 and 7 are for the part located toward one end of the line between the global systems. The variance based and uniform weightings in tables 6 and 7 shows a significant difference: the variance based weighting, taking advantage of the smaller uncertainty at close range has smaller resulting uncertainties than the uniformly weighted solution. Comparing tables 5, 6 and 7 with table 4 shows that using two global measurement systems provides improvement in the uncertainty, notwithstanding the different weighting schemes, as well as a doubling of the measurement volume!

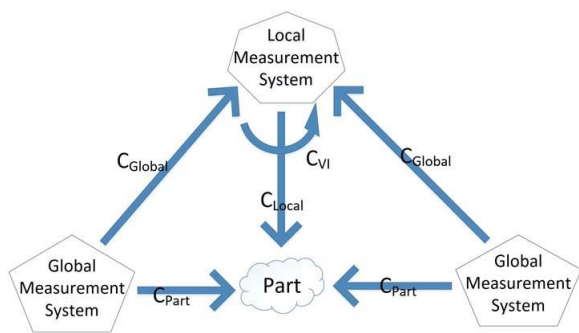


FIGURE 7. Dual Global Measurement Configuration.

TABLE 5. Typical Monte Carlo 2-Sigma Variation with Variance Based Weighting of Dual 6-Target Constellation – 25 ft and 25 ft.

	Centroid	Local Origin	Angle (μ rad)	VI
X	0.0011	0.0013	48	0.0025
Y	0.0010	0.0010		0.0010
Z	0.0011	0.0012	68	0.0032
Avg	0.0010		Avg	0.0022

TABLE 6. Typical Monte Carlo 2-Sigma Variation with Variance Based Weighting of Dual 6-Target Constellation – 5 ft and 45 ft.

	Centroid	Local Origin	Angle (μ rad)	VI
X	0.0005	0.0005	22	0.0010
Y	0.0004	0.0004		0.0004
Z	0.0005	0.0005	30	0.0013
Avg	0.0005		Avg	0.0009

TABLE 7. Typical Monte Carlo 2-Sigma Variation with Uniform Weighting ($=1$) of Dual 6-Target Constellation – 5 ft and 45 ft.

	Centroid	Local Origin	Angle (μ rad)	VI
X	0.0012	0.0014	54	0.0026
Y	0.0013	0.0013		0.0013
Z	0.0011	0.0013	75	0.0033
Avg	0.0012		Avg	0.0024

Graphs of Monte Carlo generated uncertainties for the two different weighing schemes are shown in figures 8 and 9 for ranges between 5 and 45 feet. The figures have significantly difference shapes: the variance based weighting, taking advantage of the smaller uncertainty at close range has smaller resulting uncertainties near the ends, while the uniformly weighted solution has its “sweet spot” at the middle of the measurement volume. The weighted least squares is probably the more appropriate of the two because of the large variation in standard deviations for the two global measurement systems [5]. In either case, the VI is the dominant uncertainty contributor. Had the offset between the local system origin and part been larger, the contribution would have been larger. Had the constellation been larger the pointing of the LOS vector would have been more accurate and VI uncertainty contribution smaller. It is clear that the contribution from the VI is a major factor which needs to be accounted for.

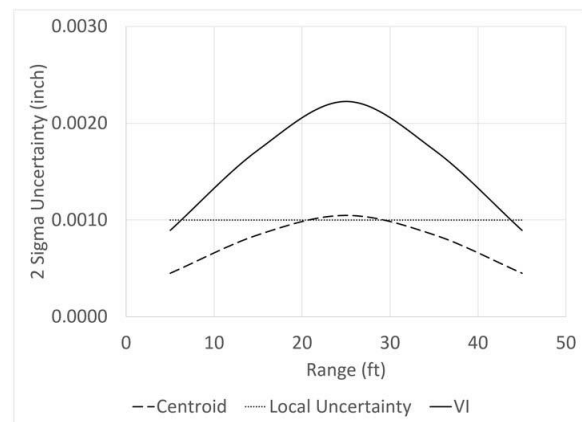


FIGURE 8. 6-Target Constellation 2-Sigma Uncertainties with Variance Based Weighting of Dual Global Measurements: Centroid, Local Origin and Virtual Instrument.

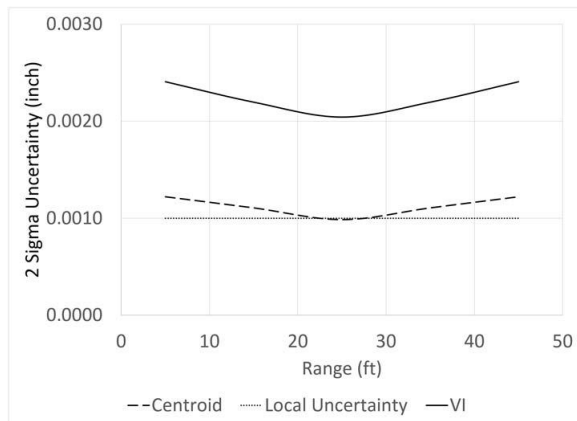


FIGURE 9. 6-Target Constellation 2-Sigma Uncertainties with Uniform Weighting (=1) of Dual Global Measurements: Centroid, Local Origin and Virtual Instrument.

CONCLUSION

Analysis shows that the virtual instrument is a major contributor to the total measurement uncertainty. When measuring a constellation of targets with a global measurement system, the angular variation is akin to pointing a gimbal can be thought of as another instrument in the chain of uncertainty: a virtual instrument. The virtual instrument uncertainty is a function of the uncertainty of the constellation measurements, number and configuration of constellation targets measured, and distance to the constellation centroid to the object. Decreasing the offset between the local system origin and part decreases the VI uncertainty. Making the constellation been larger improves the pointing of the LOS vector and makes VI uncertainty contribution smaller. Clearly, the offset between the local system origin and part is a critical element of equipment design. Monte Carlo analysis is a good way to model different VI scenarios used to evaluate several virtual instrument scenarios.

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From STEP to QIF: Product and Manufacturing Information

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ABSTRACT

For many modern companies, employees and information can be considered their most valuable resources. For a manufacturing company, Product and Manufacturing Information (PMI) can be the most valuable type of information, and the prevention of data loss is critical to the integrity of the products produced. Data translation is one of the major sources of data loss throughout the product lifecycle, and integrating product lifecycle data formats can be the solution to eliminate data loss throughout data translation. The most comprehensive solution to integrate product lifecycle data formats so far is ISO 10303, which is well known as STandard for the Exchange of Product model data (STEP). In concert with being comprehensive, STEP is a very large series of standards, which can be unwieldy at times. A newer standard effort, the Quality Information Framework (QIF), has been initiated to focus primarily on the activities associated with metrology and quality control. This standard defines a set of Extensible Markup Language (XML) schemas to resolve interoperability problems in the flow of data through quality (metrology) systems. Both STEP and QIF attempt to support the two main Geometric Dimensioning and Tolerancing (GD&T) philosophies, which are represented in the ASME Y14.5 standard and various ISO Geometric Product Specification (GPS) standards. This paper investigates how effective STEP and QIF have been in their efforts to support GD&T standards, and reports on the challenges raised by differences in the standards when trying to eliminate PMI loss during data translation.

INTRODUCTION

Product manufacturing has been very competitive in the 21st century and it has to be "faster, cheaper, and better" to compete with other products. New Concepts such as Product Lifecycle Management (PLM), Product Data Management (PDM), and Digital Enterprise (DE) have emerged to accelerate the manufacturing

process and make it more efficient. State-of-the-art products are very complicated and in many cases, there is a global supply chain that requires exchanging product digital information. Product design, manufacturing and inspection have been digitized (DE) and this digital information needs to be transferred via World Wide Web (WWW) to share them. Companies are looking for smart ways to archive old products' information and retrieve them whenever they are needed. Every company wants to digitize their product information and they do this in their proprietary format and send this information to other companies where they spend their resources (time and money) to translate – while it does not add any value to their product – this information to their own proprietary format. A typical manufacturing process is composed of Computer Aided Design (CAD), Computer Aided Manufacturing (CAM), Computer Aided Engineering (CAE), Computer Aided Inspection (CAI), Compute Aided Process Control (CAPC), etc. Product data needs to be transferred and probably translated between any of these activities, collectively known as CAx. The automotive industry reports that they spend over \$600 million annually for data translation [1]. In addition, Companies are looking for flexibility in choosing their manufacturing or inspection equipment (not a single vendor) while at the same time avoiding interoperability issues. For instance, automobile manufacturers may wish to purchase different Coordinate Measuring Machine (CMM) analysis software (proprietary data format) and CMM controller hardware (proprietary data format) to inspect their product and benefit from advantages of each brand or model.

All those problems including data translation expenses and data loss issues can be mitigated by integrating product data throughout the product lifecycle. Many efforts have been made to integrate these data formats, but most of them have only addressed the product lifecycle partially. The Initial Graphic Exchange Specification (IGES), Dimensional Metrology

Interface Standard (DMIS), Inspection plus-plus Dimensional Measurement Equipment (I++DME), Dimensional Markup Language (DML), Quality Information Framework (QIF), Standard ASCII Text (SAT), and Q-DAS are just some of these efforts.

Product and Manufacturing Information (PMI) is one a principal core data type throughout the product lifecycle, as it is required in many of the steps of product lifecycle. PMI is required in design, manufacturing, and inspection to examine if the product conforms to design specifications. PMI mostly includes Geometric Dimensioning and Tolerancing (GD&T) standards, surface finish, material properties, welding information, and 3D annotations.

ASME Y14.5M AND ISO

American Society of Mechanical Engineers (ASME) have several dimensioning standards, but ASME Y14.5 captures the majority of GD&T requirements. Unlike ASME, The International Organization of Standards (ISO) has many brief standards related to dimensioning and tolerancing, which requires users to purchase them all and understand their interrelationships. Some of the ISO GD&T standards related to Geometrical Product Specification (GPS) are ISO 8015 (Fundamentals - Concepts, principles and rules), ISO 1101 (Geometrical Tolerancing-Tolerances of form, orientation, location, and runout), ISO 5459 (Geometrical Tolerancing - Datums and datum system for geometrical tolerances), ISO 5458 (Positional tolerancing), etc. ASME and ISO can be compared in four categories: Principles, Interpretations, Symbols, and Terminology. ASME and ISO are very similar standards and they both enable the design engineers to specify the product geometry and function as desired. They both define size, form, orientation, location, profile and runout tolerances. Many GD&T symbols are common between them and they have the similar interpretations. For instance, datum feature, free state, basic dimension, projected tolerance zone, and datum target are common symbols between ISO and ASME and they have the same interpretation. Differences between these two standards are more emphasized in the upcoming sections.

Principles

ASME and ISO have fundamental differences in governing principles. In ASME, size controls form (this is called rule #1, envelope rule, or Taylor principle), but size is independent of form

in ISO standard. According to rule #1, “no variation in form is permitted if the feature is produced at its MMC limit of size” unless otherwise specified. One such exception is when using a straightness tolerance applied to the axis of a cylindrical feature. In this case, rule #1 no longer holds, and the feature can exceed the perfect-form MMC boundary. ASME provides the $\textcircled{1}$ (Independency symbol) to override rule one and ISO uses $\textcircled{E}$ (Envelope symbol) to apply rule one if required. Rule #2 of ASME (version 2009) states that regardless of feature size (RFS) applies to a feature without any material modifier, i.e. it is the default. ASME rule #2 coincides with ISO (version 2012) practice. This practice provides the same default condition to other (position, location) tolerances as applies to form tolerances – which are always evaluated at RFS [2,3].

Interpretations

ASME and ISO provide different interpretations for some geometric tolerances with the same symbol, tolerance value, and material conditions. The flatness tolerance in ASME (Y14.5 M-1994) can only be applied to one surface, while in ISO, it can be applied to two different surfaces (1101-2012) and “CZ” symbol should be used to control flatness (in this case co-planarity).

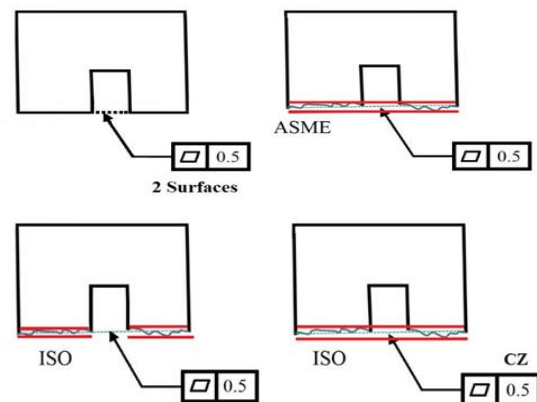


Figure 1 Different ISO and ASME Flatness Interpretations

Orientation tolerances applied to axes or median planes have a different interpretation in ISO and ASME. ISO applies the orientation tolerance to the extracted (actual) axis, line or median surface, but ASME applies to a perfect-form feature axis or plane (that of the related actual mating envelope).

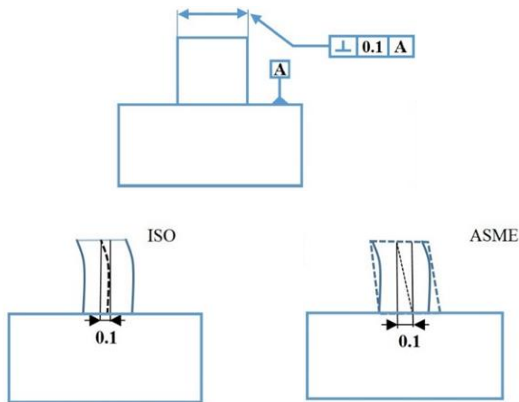


Figure 2 Comparing ASME and ISO orientation tolerances

In addition, ASME does not allow adding “Maximum Material Condition” and “Least Material Condition” modifiers to concentricity and symmetry tolerances, but ISO does. ISO applies the symmetry and concentricity tolerances to the “extracted median line” or “extracted median surface”, where ASME (2009) applies it to the median points. ISO allows the position tolerance to be applied to a surface, but ASME requires the position to be only applied to a Feature of Size (FOS). Fig. 3 illustrates applying a position tolerance to a surface.

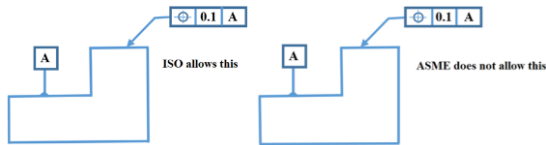


Figure 3 Applying Position tolerance to a surface (not FOS)

Run-out tolerances zone in ISO can be normal (default direction) or non-normal to the nominal surface of the part, but in ASME, the tolerance zone is always normal to the nominal profile. For Profile tolerances, ISO and ASME define a different tolerance zone at sharp corners. For ISO, the tolerance zone is defined by two equally or unequally disposed surfaces or lines formed by sweeping a sphere or a circle around the nominal profile respectively, while ASME defines two equally or unequally disposed surfaces or lines about the true profile that extend to intersection points. Composite Tolerancing: Composite tolerancing is fully described in ASME and it is used for positional and profile tolerances, but it is not defined in ISO (a composite tolerance just means two independent tolerances for ISO).

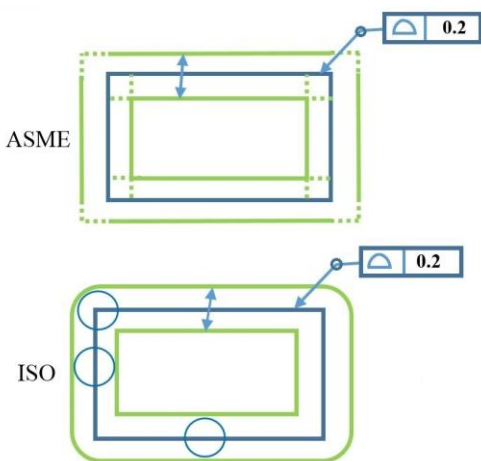


Figure 4 ISO and ASME Profile Tolerances Comparison

Symbols

There are many GD&T symbols that are defined in ISO, and not in ASME, and vice versa. These symbols and their meaning, plus their support in QIF and STEP are listed in the following tables. It is enough for a symbol to be in this table if it is defined in any versions of ASME or ISO.

Table 1 ASME and ISO Common Symbols

Symbol For	Symbols	QIF	STEP
Form Tolerances		☒	☒
Orientation Tolerances		☒	☒
Location Tolerances		☒	☒
Profile Tolerances		☒	☒
Run-out Tolerances		☒	☒
Modifying Symbols		☒	☒

Table 2 ASME Proprietary Symbols

Description	Symbols	QIF	STEP
Modifying Symbols		NO AVG	☒
Dimensioning Symbols		☒	☒

Table 3 ISO Proprietary Symbols

Description	Symbols	QIF	STEP
ISO Additional Symbols		☒	No → (A)

Terminology

ASME and ISO standards use different terminology and some of these different terminologies – although not all of them – are collected in the following table. The ASME terminology is the used throughout this paper.

Table 4 ASME and ISO terminology comparison

ASME	ISO
Feature Control Frame	Tolerance Frame
Basic Dimension	Theoretically Exact Dimension (TED)
Reference Dimension	Auxiliary Dimension
True Position	Theoretically Exact Position (TEP)
Inner Boundary	-
Outer Boundary	-
Circularity	Roundness

WHAT ARE STEP AND QIF?

STEP and QIF are both neutral data exchange formats. STEP has a broader domain than QIF, and while STEP covers whole product lifecycle QIF just covers quality (metrology) systems. QIF uses a modern information modeling language, Extensible Markup Language Schema Definition (XSD); STEP uses EXPRESS modeling language for schema definition (Table 5).

```

<xs:element name="Contact">
  <xs:complexType>
    <xs:sequence>
      <xs:element name="Name" type="xs:string"/>
      <xs:element name="FamilyName" type="xs:string"/>
      <xs:element name="Address" type="xs:string"/>
    </xs:sequence>
  </xs:complexType>
</xs:element>

```

Fragment of an XML schema definition

SCHEMA Contact_schema

ENTITY Contact

Name: STRING;

FamilyName: STRING;

Address: STRING;

END_ENTITY

END_SCHEMA

Fragment of an EXPRESS schema definition

Figure 5 XSD and EXPRESS Languages Comparison

STEP has many Application Protocols (APs), representing different parts of the product lifecycle, such as AP203, AP238, AP219, AP203 Ed2, and AP242. The first and most successful STEP AP is AP203, which did not support PMI. The second version of AP203 (AP203 Ed2)

supported PMI, but it has not been widely implemented by software vendors because it did not support many GD&T concepts. AP242 (first version), as the latest version of product model-based definition STEP AP, is used in this research to compare its PMI support with QIF's PMI support. AP242 is the most comprehensive product MBD of STEP and it covers domains that its ancestors (i.e. AP203 and AP214) did not. Unlike STEP AP242 using a single EXPRESS schema, QIF categorizes metrology framework in to four steps: product definition, measurement planning, measurement execution, and results analysis and reporting. QIF defines one or more XSD files [4] for each of these steps plus a QIF library, a common core to the quality framework (Figure 5).

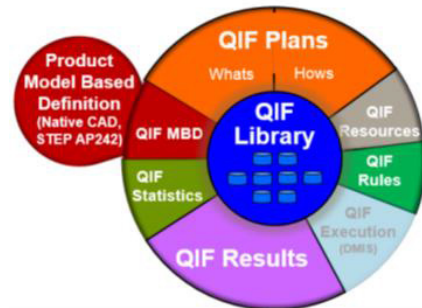


Figure 5 QIF Application Schemas and Library [5]

PMI IN STEP AND QIF

QIF V2.1 models ASME Y14.5M (Version 1982, 1994, 2009), ISO 1101 (Version 1983, 2004, 2012), BS_8888_2004, JIS, DIN, ASME Y14.41-2003, ASME Y14.36-1996, ASME Y14.6-2001. STEP AP242 models ISO (Different Versions), ASME (Version 1994, 2009), ASME Y14.41-2003, and ISO 16792. PMI in QIF are defined in schemas: "Characteristics," "IntermediatesPMI," and "PrimitivesPMI". The Characteristics schema defines the upper level PMI such as dimensional and geometric tolerances including size, form, orientation, profile, runout, and location tolerances. IntermediatesPMI includes mid-level PMI such as Datum Translation, Degrees of Freedom, Datum Precedence, and different tolerance zones. PrimitivesPMI models lower-level PMI including – but not limited to – different manufacturing methods, internal-external features, and the number of PMI dimensions (two or three-dimensional features or characteristics). In STEP, PMI are defined in a single EXPRESS Schema file. QIF is a feature-and characteristic-based data format that defines four aspects of a characteristic or

characteristic: *Definition*, *Nominal*, *Item*, and *Actual*. This allows QIF to define PMI and report measurement results, whereas STEP just defines the PMI. QIF uses a hierarchy structure to define PMI, which makes it "smarter" than STEP. STEP uses a pool of entities to connect to other pools to define the PMI. For instance, the all-over symbol (Figure 6) is applied through referencing a *product_definition_shape* entity by a *geometric_tolerance* entity, which can be any kind of geometric tolerance such as form or orientation tolerances. The semantics of this referencing is then checked by quality software (the all-over symbol can only be used for profile tolerance, e.g.). However, according to QIF, all-over symbol is defined in "ExtentType" complex type, which is referenced in the "SurfaceProfileCharacteristicDefinitionType". In this case, the smart hierarchy does not allow the all-over symbol to be used for tolerances other than surface profile unless there is an element inside a complex type that references "ExtentType" complex type (Fig. 6).

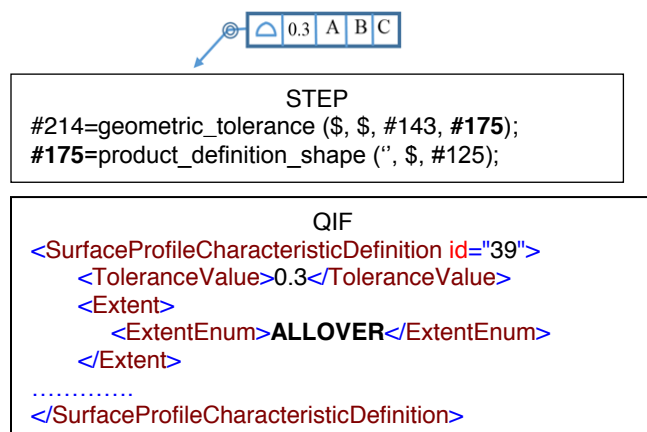


Figure 6 Comparing all-over symbol (ASME Y14.5- 2009, section 8.3.1.6) definition in QIF and STEP

In addressing ISO and ASME differences (Principles, Interpretations, Symbols, and Terminology), QIF and STEP allow explicit reference to the GD&T standard (ISO or ASME and their version) for clarification purposes. The principles and interpretation used in a drawing depend on the referenced standard, and it is up to the end user or the software to interpret them for the intended use (including the part passing or failing inspection). In terms of symbols, QIF and STEP may or may not model some of the GD&T symbols (collected in table 1, 2, and 3). Regarding terminology, QIF and STEP use

ASME and ISO terminology as their preferred terminology respectively, unless a concept is just defined either in ASME or in ISO.

QIF and STEP both include size, form, orientation, location, runout, and profile tolerances, and many of their proper modifiers (i.e. statistical tolerance, free state, etc.). They both model datum feature definition (including multiple datum feature), datum modifiers (e.g. translation, basic, etc.), material modifiers, datum targets, tolerance classes, and composite tolerances. QIF and STEP use different methods to associate the feature of size (FOS) with the corresponding feature control frame (Fig. 7 and Fig 8). QIF defines features (cylinder or hole in this case), and the corresponding FOS id is referenced inside the characteristics. The STEP method is that the corresponding geometric tolerance (position tolerance in Fig. 7 and Fig. 8) entity references a "dimensional_size" (hole diameter in this case) or "dimensional_location" entity that defines FOS [6].

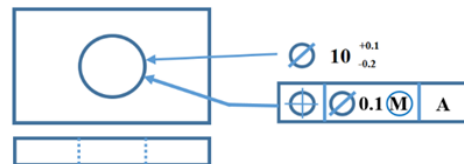


Figure 7 Associating a position tolerance with a (hole) diameter dimension

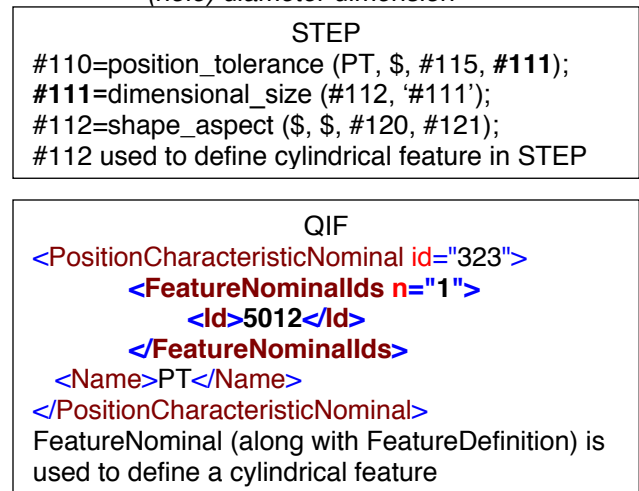


Figure 8 Comparing STEP and ISO methods in associating FOS with a (position) tolerance

Dimensional tolerances (depending on type) can have different modifiers such as Continuous Feature (CF), Controlled Radius, Statistical Tolerance, etc. QIF lacks modeling dimensional

tolerance modifiers in this regards. For instance, dimensional tolerances in QIF do not include Area Diameter, and Volume Diameter modifiers. STEP incorporates dimensional tolerances modifiers better than ASME (including the aforementioned modifiers).

In terms of different tolerance zones, STEP gives a pool of choices and flexibility for the user by defining tolerance zones, but QIF defines a hierarchy based tolerance zones based on geometric tolerance type. For instance, QIF defines Spherical, Diametrical, and Non-Diametrical zones for concentricity tolerance. In addition, both standards support unit-basis tolerance zones (e.g. tolerance per length, area, or arc length).

QIF models surface finish parameters according to ASME Y14.36-1996, while STEP does not model any surface finish standard. Knurling information is part of PMI that neither QIF nor STEP supports at this time. Part of product surface may receive additional treatment such as shot peen and induction hardening, that it is not modeled neither in STEP nor in QIF.

3D annotation is an important part of PMI that both STEP and QIF define ways to represent them. Presentation of PMI data has been standardized in ASME Y14.41-2003 and ISO 16792. STEP models the aforementioned standards, but QIF only supports ASME Y14.41-2003. In QIF, the Visualization schema contains the information (ASME Y14.41-2003) required to represent different GD&T, annotations and saved views.

Three ASME Y14.5M-2009 symbols (Spotface, Counterbore, and Countersink) can require machining features rather measurement features. According to AP242 documentation, AP242 utilizes machining features from other APs (AP238), but these are not supported in first version of AP242. QIF defines its own measurement features, including twenty-nine features that can be used instead of machining features.

Conclusion

This paper presented an overview of PMI modeling in QIF V2.1 and STEP AP242 (First Edition) standards. It has been shown that both standards are able to capture PMI concepts, especially ASME and ISO GD&T philosophies. STEP and QIF standards do not model a few GD&T symbols, manufacturing information, and surface finish in the case of STEP. As the standards grow and become more mature, it is expected that information loss in data translation

due to lack of standard modeling methods will continue to be reduced.

Acknowledgments

The authors would like to acknowledge QIF working groups for their efforts in developing the QIF standard and Center for Precision Metrology (CPM) affiliate's research grant.

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A ROUNDNESS MEASUREMENT WITHOUT OMISSION OF HARMONIC COMPONENTS USING DATA INTERPOLATION COMBINED WITH SEQUENTIAL THREE-POINT METHOD

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INTRODUCTIONS

The sequential three-point method is a useful technique for on-machine measurement of roundness especially at the factory site because of its simple and easy calculation algorithm. A distinctive feature of the method is that the error motion of measuring equipment can be eliminated. On the other hand, the detectable order of harmonic components is limited by the spacing angle between sensors. The smaller the spacing angle is, the higher order the harmonic component can be detected. However, in practical measuring equipment, the size of the sensors restricts the trial to minimize the spacing angle.

In this study, to extend the detectable order of harmonic component to higher region, an interpolation method of output data of the displacement sensors combined with the sequential three-point method is proposed. And the effectiveness of the method is confirmed by numerical simulations and measuring experiments.

However the method has still a disadvantage that some specified harmonic components are omitted depending on the setting angle between the sensors.

PRINCIPLE OF THREE-POINT METHOD FOR ROUNDNESS MEASUREMENT [1][2]

The arrangement of test piece and three displacement sensors is shown in Figure 1. The sensor #2 is aligned so that the axis of which passes through the center of rotation O of the test piece. The sensors #2 and #3 are aligned on both side of the sensor #1 with the spacing angle φ , respectively. The rotational angle of the test piece is indicated by θ .

By rotating the test piece on rotary table, discontinuous radial figure data are obtained with the

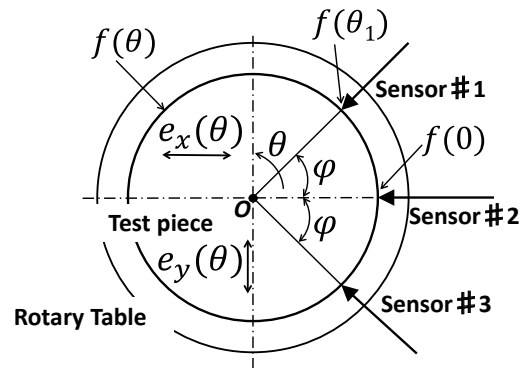


Figure 1 Arrangement of sensors in sequential three-point method for roundness measurement

sensors at equal angular interval of φ along a circumference line of the test piece as shown in Figure 1. Supposing that $f(\theta_i)$ and $e_x(\theta_i)$ and $e_y(\theta_i)$ are the discontinuous radial figure of the test piece and the error motions of the test piece in x and y directions, respectively, the output signals of the three sensors $m_1(\theta_i)$, $m_2(\theta_i)$, $m_3(\theta_i)$ are expressed as Eqs.(1) to (3).

$$m_1(\theta_i) = f(\theta_i - \varphi) + e_x(\theta_i) \cos \varphi - e_y(\theta_i) \sin \varphi \quad (1)$$

$$m_2(\theta_i) = f(\theta_i) + e_x(\theta_i) \quad (2)$$

$$m_3(\theta_i) = f(\theta_i + \varphi) + e_x(\theta_i) \cos \varphi + e_y(\theta_i) \sin \varphi \quad (3)$$

To eliminate the error motion $e_x(\theta_i)$ and $e_y(\theta_i)$, the differential output signals are calculated using Eqs.(4) and (5).

$$\mu_1(\theta_i) = m_1(\theta_i) - m_2(\theta_i) \cos \varphi \quad (4)$$

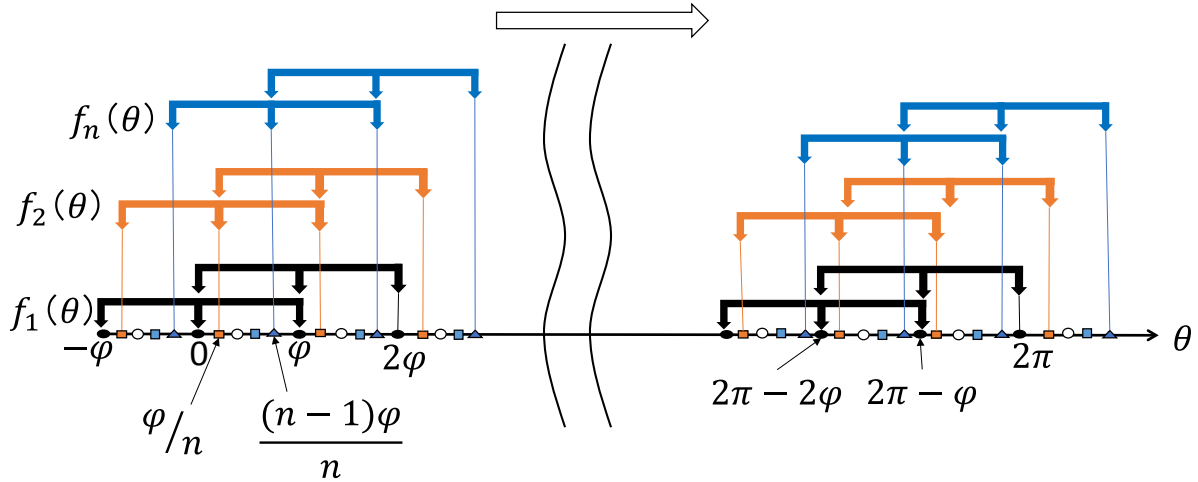


Figure 2. Concept of roundness measurement by interpolation combined with sequential three-point method expressed in rectangular coordinates

$$\mu_2(\theta_i) = m_2(\theta_i) \cos \varphi - m_3(\theta_i) \quad (5)$$

The second difference of the output signals $\Delta\mu(\theta_i)$ is expressed by Eq.(6).

$$\begin{aligned} \Delta\mu(\theta_i) &= \mu_1(\theta_i) - \mu_2(\theta_i) \\ &= f(\theta_i - \varphi) - 2f(\theta_i) \cos \varphi + f(\theta_i + \varphi) \end{aligned} \quad (6)$$

Rewriting the Eq. (6), The recurrence formula of the radial figure is obtained as Eq.(7).

$$f(\theta_i + \varphi) = \Delta\mu(\theta_i) + 2f(\theta_i) \cos \varphi - f(\theta_i - \varphi) \quad (7)$$

Eq.(7) shows that the radial figure $f(\theta_i + \varphi)$ can be calculated using $f(\theta_i)$ and $f(\theta_i - \varphi)$ with the second difference of output signals of the sensors #1 to #3. To solve successively the recurrence formula, two initial values of radial figure, $f(\theta_0)$ and $f(\theta_1)$ are necessary. When the sensors #1 and #2 are at the initial start position, $\theta=0$ and φ , respectively, $f(\theta_0)$ and $f(\theta_1)$ are average radius of the test piece and the term corresponding to eccentricity. In roundness measurement, both of them are considered to be 0 [3].

INTERPOLATION METHOD PROPOSED [3]

The feature of the method proposed is to interpolate the radial figure data obtained at every φ/n steps between the data obtained at equal angular interval of φ along a circumference line of the test piece as shown in Figures 2 and 3. In practice, the necessary data at every φ/n steps

are used from the continuous data collected by three sensors. The procedures of the interpolation method are explained as follows.

- Step 1 Obtain the discontinuous radial figure data within one revolution $f_0(\theta_i)$ when the starting position θ_0 is 0, where $i=0,1, \dots, 2\pi/\varphi - 1$. The figure data is expressed by straight line graph at every φ steps as shown in Figure 3.
- Step 2 Obtain the straight line graphs of radial figure data $f_1(\theta_i + \varphi/n)$, $f_2(\theta_i + 2\varphi/n)$, $\dots$, $f_j(\theta_i + j\varphi/n)$, where $j=1,2, \dots, n-1$. After the procedure, n line graphs are obtained with shifting the starting position by every φ/n steps.
- Step 3 Suppose that $f_0(\theta_i)$ is the temporary standard for linear interpolation of radial figure profile $f_j(\theta_i + j\varphi/n)$.
- Step 4 Align the phases of $f_j(\theta_i + j\varphi/n)$ same as that of $f_0(\theta_i)$ and estimate the difference $E_j(\theta_i + j\varphi/n)$ between the values of $f_j(\theta_i + j\varphi/n)$ and the temporary standard at $\theta = \theta_i + j\varphi/n$. $E_j(\theta_i + j\varphi/n)$ is expressed by Eq(8).

$$\begin{aligned} E_j(\theta_i + j\varphi/n) \\ = f_0(\theta_j) + \frac{f_0(\theta_i + \varphi) - f_0(\theta_i)}{n} \times j - f_j(\theta_i + j\varphi/n) \end{aligned} \quad (8)$$

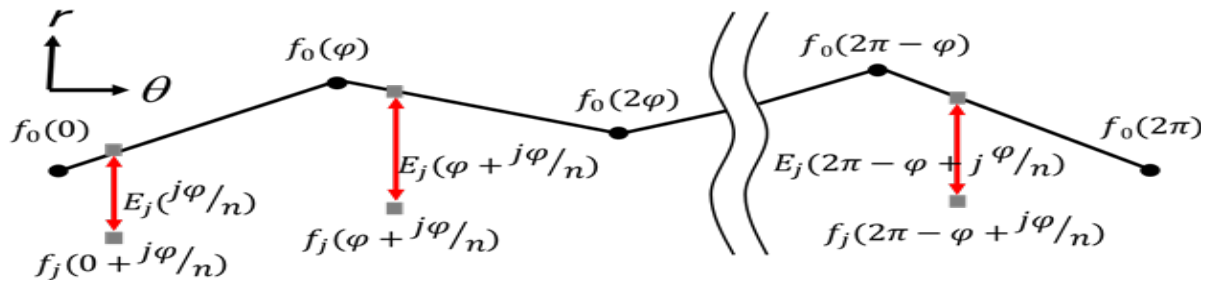


Figure 3 Composition of figure profile by interpolation

Step 5 Supposing that $f_j(\theta_i + j\varphi/n)$ is expressed as $f_0(\theta_i + j\varphi/n) + a_j(\theta_i + j\varphi/n) + b_j$, calculate the correlation coefficients a_j and offset b_j by a least square method. $f_j(\theta_i + j\varphi/n)$ is expressed by Eq(9).

$$f_j(\theta_i + j/nx) = f_0(\theta_i + j/nx) + a_i(\theta_i + j/nx) + b_j \quad (9)$$

Step 6 Compose the radial figure profiles calculated at step 5.

Step 7 Finally, eliminate the 1st harmonic component, which is corresponding to the eccentricity, from the radial figure profile of the combined.

NUMERICAL SIMULATION OF ROUNDNESS MEASUREMENT BY THE METHOD PROPOSED

To confirm the validity of the algorithm proposed, numerical simulations of roundness measurement were carried out. φ and n were supposed to be $\pi/6$ and 6, respectively. Figure 4 shows the figure profile and error motion of the model used in the simulation. Red line shows the figure profile of the model. The model has characteristic 10th harmonic component with $1\mu\text{m}$ amplitude. Blue dotted line shows the error motion of the test piece. That includes 7th and 8th harmonic components and eccentric component. Figure 5 shows the result of the measuring simulation by conventional sequential three-point method. The error motion of the test piece is eliminated. However, the 10th harmonic component is omitted. On the other hand, Figure 6 shows the result of the measuring simulation by improved sequential three-point method proposed. The 10th harmonic component is detected with high fidelity and the error motion of the test piece is clearly eliminated. The other simulations showed that the 7th to 11th harmonic components were measured without

any decrease of amplitude by the improved sequential three-point method proposed.

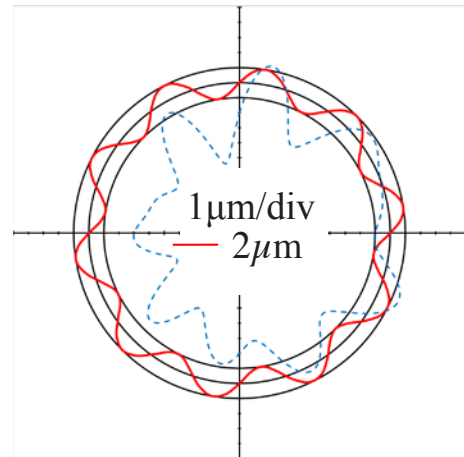


Figure 4 Figure profile of the test piece (red line) and error motion (blue dotted line) of the test piece used in the simulation

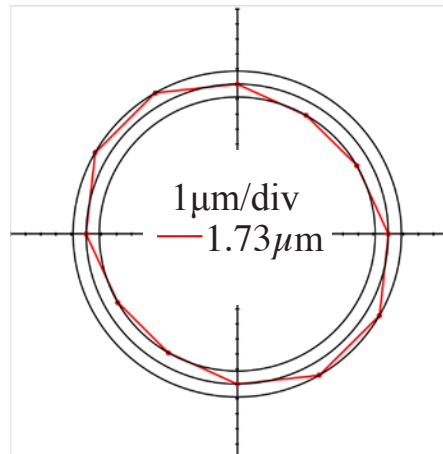


Figure 5 Result of measuring simulation by conventional sequential three-point method

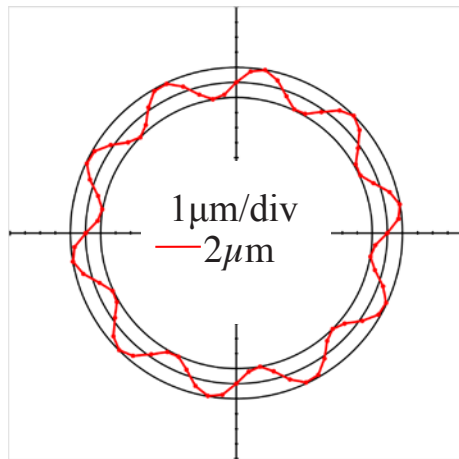


Figure 6 Result of measuring simulation by improved sequential three point method proposed

MEASURING EXPERIMENT

To ascertain the principle described above, measuring experiment was carried out. The experimental measuring equipment was assembled using a rotary table with a hydrostatic air bearing and three capacitance type displacement sensors. ϕ and n were set to be $\pi/6$ and 30, respectively. The spacing angle and rotating angle were measured by a potentiometer built into the rotary table. The location of test piece on the rotary table was adjusted so that the eccentricity measured by the sensor #1 showed the minimum value. The circular tube finished by turning was used as the test piece. A 3rd harmonic component was produced on the work piece surface by fastening the 3-jaw chuck of the lathe in turning. The radial figure profile of the test piece was measured by the method proposed. Figure 7 shows the result of the measurement. The characteristic 3rd harmonic component was observed. The roundness was $3.38\mu\text{m}$. Figure 8 shows the figure profile of the identical test piece measured by a commercial roundness measuring machine (Taylor Hobson TALYROND 290RSU). The roundness was $3.69\mu\text{m}$. Both of the roundness value and figure profile in Figure 7 coincide with that in Figure 8. However the method has still a peculiar disadvantage that some specific harmonic components are omitted depending on the setting angle between the sensors. Another compensation method is required to improve the disadvantage.

SUMMARY

The new concept of interpolation method combining with a sequential three-point method is proposed to extend the detectable order of harmonic

component to higher region in roundness measurement. The validity of the method is confirmed by a numerical simulation and a measuring experiment.

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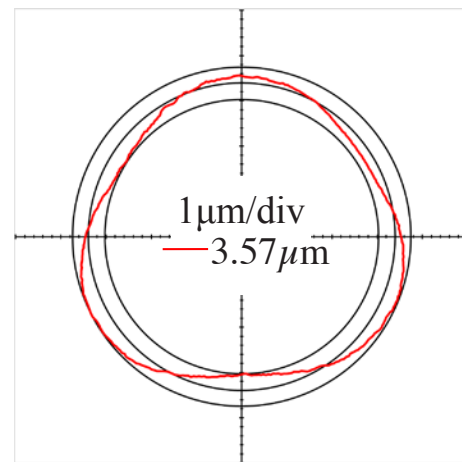


Figure 7 Radial figure profile of the test piece measured by the method proposed

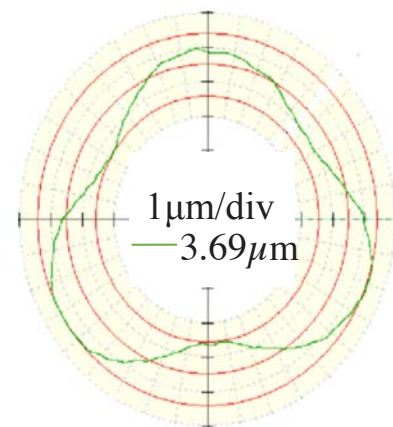


Figure 8 Radial figure profile of the test piece measured by a commercial measuring machine

DEVELOPMENT OF WHITE LIGHT SCANNING INTERFEROMETRY FOR FREEFORM SURFACE METROLOGY

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CURVATURE METHOD FOR FREEFORM SURFACE METROLOGY

Freeform surfaces may be classified as complex geometrical features which have no invariance degree. The term 'freeform' is generally used for surfaces without any rotational symmetry. [1] A variety of methods have been suggested for diffusely reflecting freeform surfaces metrology; however, it is much more difficult to measure the shape of specular freeform surfaces, since the surfaces themselves are not visible. We only see their effect from the reflection of incoming light.[2] The measurement of specular surfaces can be done by CMM (Coordinate Measuring Machine) or interferometric means, as shown in Figure 1.

However, CMM has limited measurement accuracy and typical interferometry cannot cover the large departure from the reference, flat or sphere. Null testing could be one possibility to cover insufficient dynamic range of full-aperture interferometry. However, the use of null optics not only adds cost, but also creates additional metrology problems and increases measurement uncertainty. Another option is sub-aperture testing, which involves guiding a sensor along the entire surface and then combining them into a unified profile of the test surface. However, for the sub-aperture testing, the presence of additional mechanical and optical degrees of freedom introduce unavoidable errors in mechanical alignment and motion control caused by

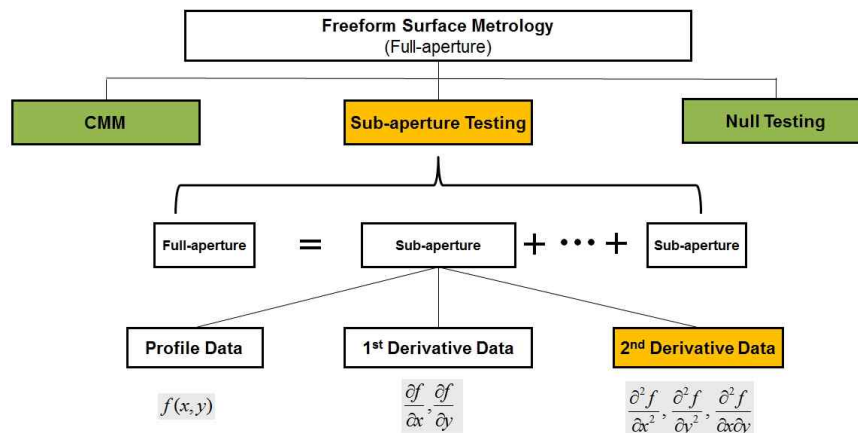


FIGURE 1. The overview of freeform surface metrology

temperature and non-orthogonality between each axis on the machine. These problems are related to the geometrical property obtained from the sub-aperture. However, the local profile or the slope variation depended on the elevation and attitude of the scanning probe, whereas the curvature, being an intrinsic property of the local profile, does not depend on the elevation or tip-tilt motion of the measuring probe, as shown in Figure 2. [3]

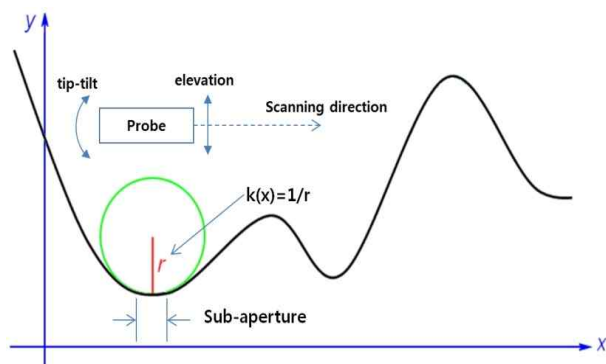


FIGURE 2. Curvature does not depend on the attitude of probe

When the profile reconstruction was simulated incorporating pre-assigned position errors for the hyperbolic profile, it showed that the curvature method was unaffected by the probe's scanning error. [4]

WHITE LIGHT SCANNING INTERFEROMETRY AS FREEFORM PROFILER

The entire surface profile was divided into a number of sub-aperture with an overlapping area. Each sub-aperture can be measured using various types of surface profiler. However, measurement accuracy depends on the type of profiler in realizing the curvature method.

A typical laser based interferometry, Fizeau or Twyman-Green, was designed for measuring the shape of surface by comparing the wavefront reflecting from the surface with the reference wavefront emitted from the interferometry. While obtaining the curvature from the sub-aperture by using typical interferometries, if the reference is an ideal plane wavefront, the curvature of the reflected wavefront depends on the distance between the curvature sensor and the surface, as shown in

Figure 3. In using these kinds of interferometry as curvature sensor, the distance between the sensor head and the surface should be controlled by additional devices, such as laser interferometer.

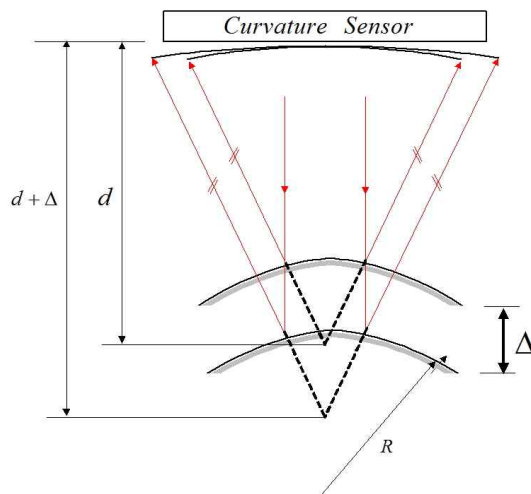


FIGURE 3. Dependence of curvature on distance between curvature sensor and surface while using a typical interferometry as curvature sensor.

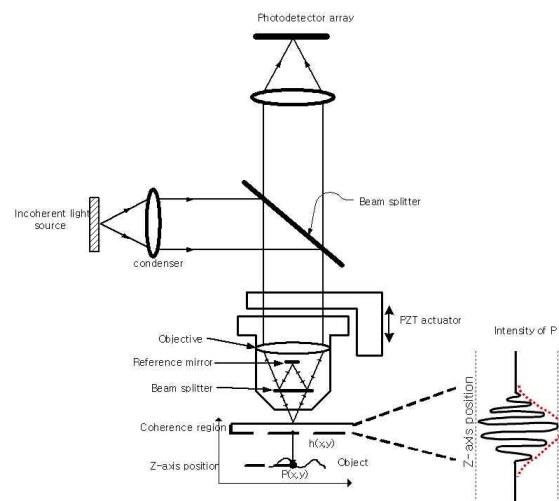


FIGURE 4. Working principle of white light scanning interferometry performed with Mirau type. Interferograms are sampled using a photodetector array while moving the target surface along z direction using a PZT actuator.

One promising possibility is using a white light scanning interferometry that moves along z direction to determine the curvature at each sub-

aperture (see Figure 4). When the target passed through the coherence region, where the optical path difference between reference arm and test arm was within the temporal coherence length of white light, interference fringes appeared in the narrow spatial region due to short coherence, being wrapped by visibility envelop profile. The envelop peak detection led to the exact position where the optical path difference was zero. At a point $p(x,y)$ on the surface, its height $h(x,y)$ was determined by detecting the PZT travel length. The curvature was determined from the surface profile $h(x,y)$ at a sub-aperture while scanning the entire surface target.

OVERVIEW OF THE DESIGN

To apply the curvature method for freeform surface metrology, a new system is currently under development, as shown in Figure 5. This system is based on the principle of white light scanning interferometry to determine sub-aperture topography, while moving the target surface along z direction using a PZT actuator. Figure 6 shows the photograph of the curvature measuring system, which is currently under development.

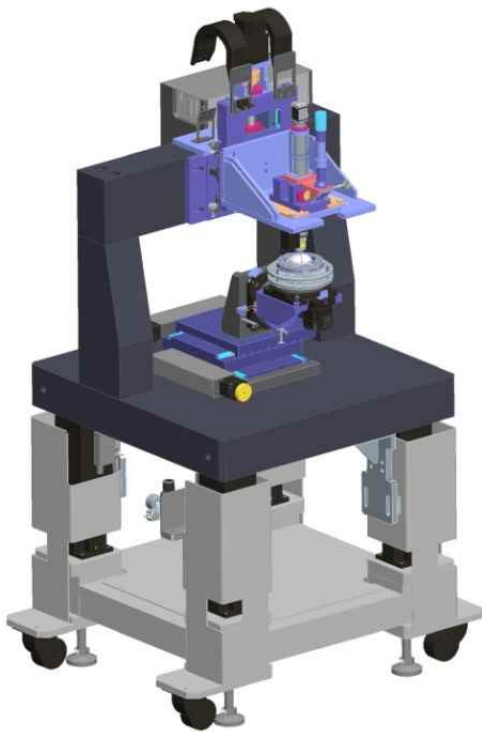


FIGURE 5. Model of the white light scanning interferometry developed for freeform surface metrology



FIGURE 6. Photograph of the measuring system under development



FIGURE 7. Rotating range -5° ~ 60° , (Right) 30°

All the axes are controlled automatically from the controller with 5 degrees of freedom, which comprises 3 translation axes and 2 rotation axes. A computer controls both the white light scanning interferometry and 5-axes motion. To keep the optical axis perpendicular to the measuring surface, the combination of 5 axes is controlled. Rotation table works within the range of up to 60 degree. The new system was installed in a temperature controlled laboratory on a vibration isolated granite table.

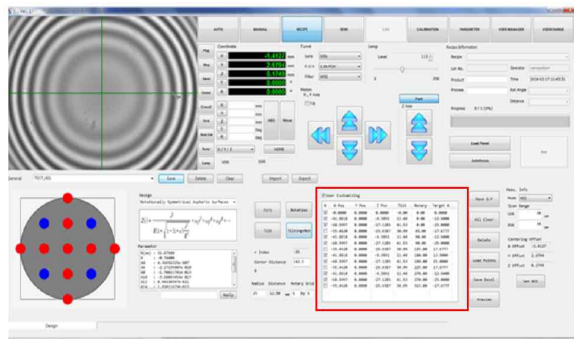


FIGURE 8. GUI consisting of several windows for control and sequential measuring process

Figure 8 shows GUI (Graphical User Interface), which consists of several windows for the control and indicator. It allows users to control the number of measuring points and obtain curvature at each sub-aperture.

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MASS LOADING COMPENSATION FOR FREQUENCY RESPONSE FUNCTION MEASUREMENTS BY INVERSE RCSA

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INTRODUCTION

Machining operations can be stable or unstable depending on the cutting parameters and machine-spindle-holder-tool assembly frequency response function (FRF). In some cases, the workpiece FRF can influence the machining behavior as well. Stable, or chatter-free, cutting conditions can be predicted using process stability models [1]. These models require the machine-spindle-holder-tool assembly FRF (and sometimes the workpiece FRF). System FRFs can be obtained by measurement (i.e., modal testing) or models (analytical and/or numerical methods). Schmitz and Donaldson [2] first presented the Receptance Coupling Substructure Analysis (RCSA) method to predict machine-spindle-holder-tool assembly FRFs by coupling individual component FRFs (or receptances). This method reduces measurement time because assembly FRFs can be predicted rather than measured. Subsequent publications have improved the technique [3-12].

In this paper a novel application of RCSA is presented in order to improve FRF measurement accuracy. In modal testing, FRFs are measured using an instrumented hammer to excite the system and (typically) an accelerometer to record the response. However, the measured FRF differs slightly from the actual FRF due to the accelerometer and cable mass. In order to compensate for this mass loading, the inverse RCSA approach is applied here.

RCSA BACKGROUND

RCSA is used to predict an assembly's receptances by coupling receptances from the individual components. The connections between components can be rigid or flexible with or without energy dissipation (damping) [13]. An example for rigid coupling of two components is displayed in Fig. 1.

For this example, the component direct receptances can be described as $h_{1a1a} = x_{1a}/f_{1a}$ (component I) and $h_{1b1b} = x_{1b}/f_{1b}$ (component II). The compatibility condition for the rigid coupling is $x_{1b} - x_{1a} = 0$. The equilibrium condition, $f_{1a} + f_{1b} = F_1$,

relates the internal (component) forces to the external (assembly) forces. The assembly (III) direct receptance, H_{11} , at assembly coordinate, X_1 , can be expressed as shown in Eq. 1.

$$H_{11} = \frac{X_1}{F_1} = h_{1a1a} - h_{1a1a} (h_{1a1a} + h_{1b1b})^{-1} h_{1a1a} \quad (1)$$

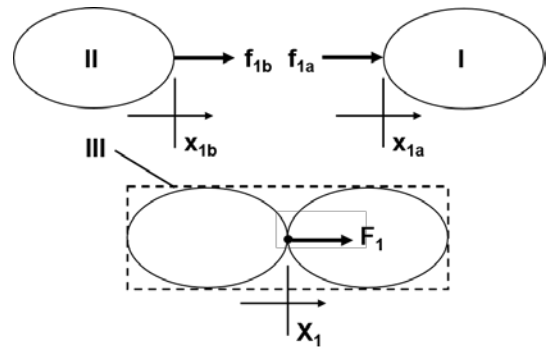


Figure 1. Two component RCSA model: I and II are individual components and III is the assembly.

INVERSE RCSA APPROACH FOR MASS LOADING COMPENSATION

A tool point FRF is measured by modal testing as shown in Fig. 2a. The experimental FRF differs from the actual FRF due to the accelerometer and cable mass. A reduction in the natural frequency(s) and FRF magnitude may be observed, depending on the amount of mass loading.

The accelerometer and cable mass can be compensated using inverse RCSA, where the corresponding RCSA model is depicted in Fig. 2b. In this model, it is assumed that accelerometer is rigidly coupled to the tool point. The measurement provides the assembly receptance, $H_{11} = X_1/F_1$. The accelerometer/cable receptance is $h_{1a1a} = x_{1a}/f_{1a}$, while the unknown tool point receptance is $h_{1b1b} = x_{1b}/f_{1b}$. The tool point receptance can be determined by rearranging Eq.1 as shown in Eq. 2. This approach is

referred to as inverse RCSA since it is a decoupling, rather than a coupling, step.

$$h_{1b1b} = -h_{1a1a} + h_{1a1a} (h_{1a1a} - H_{11})^{-1} h_{1a1a} \quad (2)$$

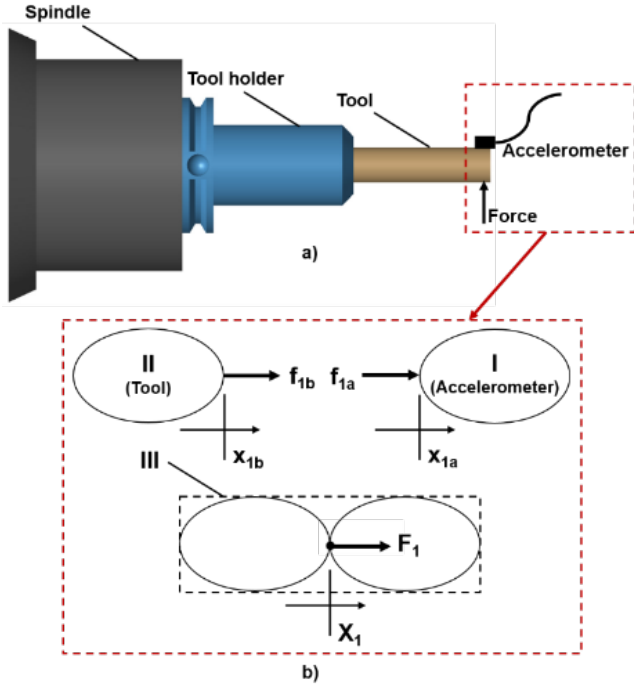


Figure 2. a) FRF measurement and b) RCSA model.

For this study, the accelerometer/cable is defined as a point mass. The corresponding receptance is provided in Eq. 3, where m is the mass and ω is the frequency (rad/s).

$$h_{1a1a} = -1/(m\omega^2) \quad (3)$$



Figure 3. 12.7 mm diameter rod setup.

RESULTS

Experiments were performed on two setups: 12.7 mm diameter and 6.35 mm diameter steel rods clamped at one end in a cantilever configuration. Figure 3 displays a photograph of the 12.7 mm diameter rod measurement platform. A split clamp was used to hold the tool at the desired cantilever length. The 6.35 mm diameter rod was clamped in an ER16 collet holder with a CAT-40 interface, which was then secured using a manual draw bolt in a spindle nose attached to a large steel block; see Fig. 4.

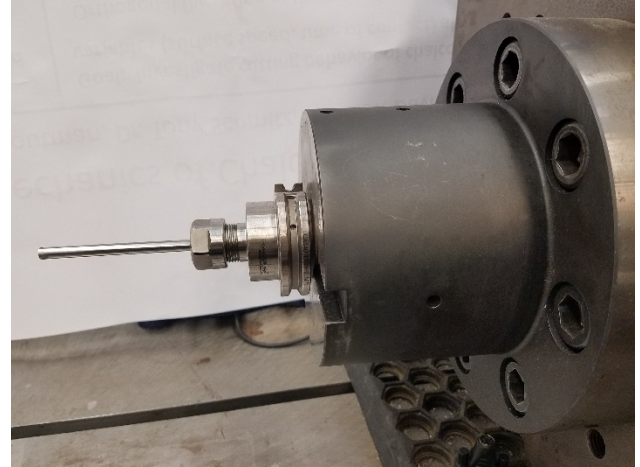


Figure 4. 6.35 mm diameter rod setup.

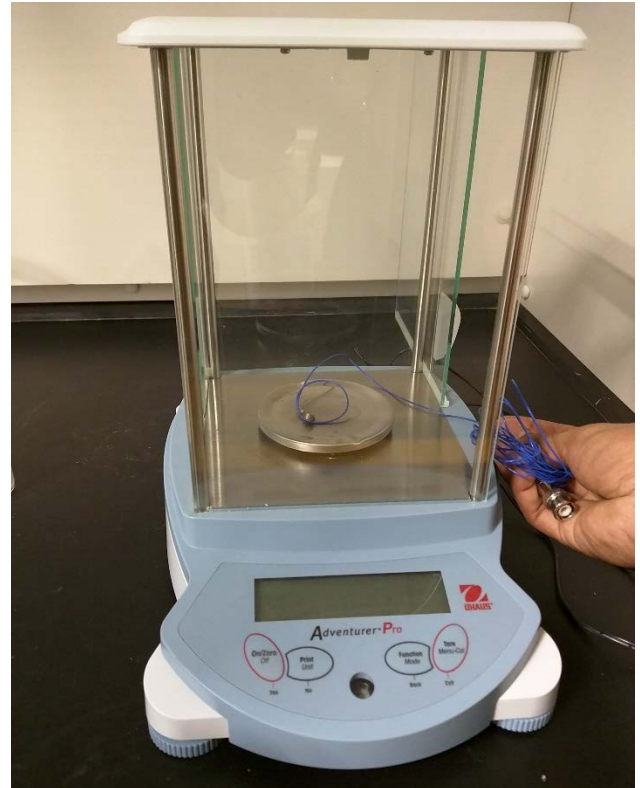


Figure 5. Accelerometer-cable mass measurement.

For both rod diameters, multiple stickout lengths were selected and measurements were performed with: a medium-size accelerometer (PCB 352A21), a small-size accelerometer (PCB 352C23), and a laser vibrometer (Polytec OFV-534). To obtain the accelerometer/cable masses, measurements were performed using an Ohaus AV264C Adventurer ProAnalytical Balance (0.1 mg resolution). A measurement example is displayed in Fig. 5, where the accelerometer, approximate catenary length of the cable from the tool point measurement (for the cable mass), and the modal wax were included.

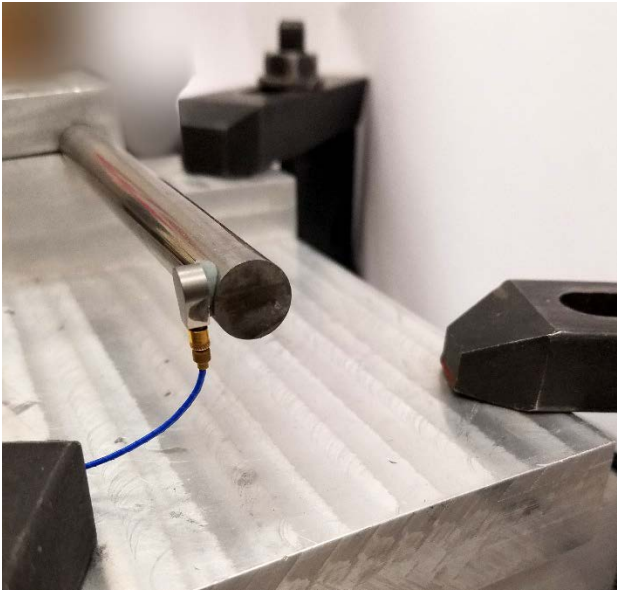


Figure 6. Medium accelerometer attached to the 12.7 mm diameter rod using modal wax.

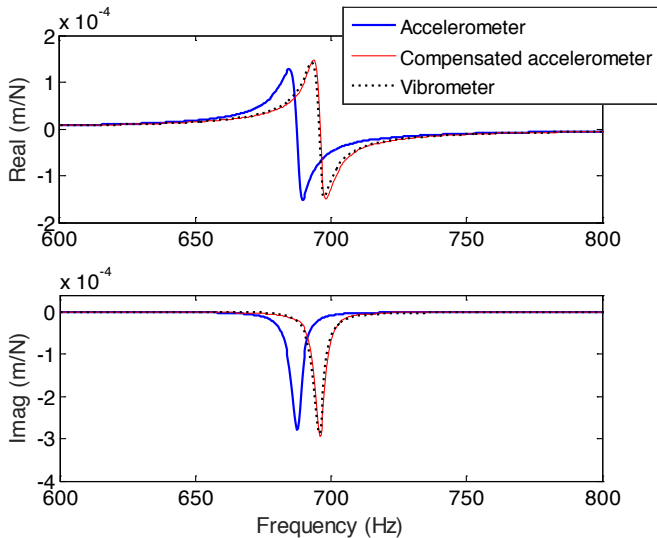


Figure 7. Results for 12.7 mm diameter rod with a stickout length of 102 mm using the medium accelerometer.

Figure 6 shows the medium accelerometer attachment for a stickout length of 102 mm for the 12.7 mm diameter rod. The measurement and compensation results are displayed in Fig. 7, where the accelerometer/cable mass was 706 mg. It is seen that the accelerometer FRF has a lower natural frequency than the vibrometer (non-contact) FRF. Using Eq. 2 and the measured mass, the mass loaded FRF is compensated to remove the mass loading effect. This result matches closely with the vibrometer result (phase error in the vibrometer measurement due to a time delay between the hammer and vibrometer electronics was compensated using the technique described in [14]).

A summary of 12.7 mm diameter rod measurement results for both accelerometers at four different stickout lengths is provided in Table 1.

Table 1. 12.7 mm diameter rod results.

Stick-out (mm)	Vibro. freq. (Hz)	Accel. freq. (Hz)	% error	Comp. freq. (Hz)	% error
Medium accelerometer (706 mg)					
102	695.4	687.4	1.10	695.8	-0.05
111	597.8	592	0.97	598.5	-0.11
118	538.3	533.3	0.92	537.5	0.15
124	492.1	487.1	1.01	492.1	0
Small accelerometer (275 mg)					
102	698.1	695	0.44	698.1	0
111	613.8	611.5	0.37	614.2	-0.06
118	555.4	552.8	0.46	555	0.07
124	503	501.3	0.33	502.8	0.03

*Stickout lengths are approximate. Different setups were used between the small and medium accelerometers.



Figure 8. Small accelerometer attached to the 6.35 mm diameter rod using modal wax.

Figure 8 shows the small accelerometer attached to the 6.35 mm diameter rod. Measurement results are presented in Table 2.

Table 2. 6.35 mm diameter rod results.

Stick-out (mm)	Vibro. freq. (Hz)	Accel. freq. (Hz)	% error	Comp. freq. (Hz)	% error
Medium accelerometer (680 mg)					
79	687.8	671.0	2.4	688.2	-0.06
89	545.5	532.2	2.4	545.9	-0.07
Small accelerometer (275 mg)					
79	688.2	646.0	6.1	688.9	-0.10
89	546.3	516.0	5.5	547	-0.12

*Stickout lengths are approximate. Different setups were used between the small and medium accelerometers.

CONCLUSIONS

This paper described the application of inverse Receptance Coupling Substructure Analysis (RCSA) to mass compensation for accelerometer-based impact testing. The measurement (assembly) FRF was used together with a point mass model for the accelerometer/cable to determine the compensated tool point FRF. Experiments were completed for two tool diameters and multiple stickout lengths for a cantilever configuration. The average percent error in fundamental natural frequency after compensation was -0.03%.

ACKNOWLEDGEMENTS

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DYNAMIC RESPONSE CHARACTERIZATION OF FORM MEASURING STYLI

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INTRODUCTION

Manufacture of precision air bearing spindles requires a metrology feedback loop that typically involves form metrology instruments for roundness and flatness. Implementing this in a production environment leads to stylus-based roundness testers at every grinder and diamond turning machine. In order to calibrate and perform regular checks of this large population of stylus systems, an artifact was designed that provides a quasi-static and dynamic fidelity standard. Amplitude attenuation as a function of frequency is demonstrated for cartridge plunger, lever head and air bearing roundness tester styli.

METHODOLOGY

A calibrated test artifact is used to characterize the frequency response of a variety of styli. The artifact shown in Figure 1 is a hardened 440C steel cylinder 100 mm diameter with a well-defined measurement band for circular flatness measurements.



FIGURE 1. Hardened steel artifact with defined measurement band for calibrating circular flatness and roundness instruments.

Before a sinusoidal fidelity form for dynamic characterization is imparted on the measurement band, the band and opposing mounting face are rotary ground flat to better than 200 nm. Then the calibration sinusoid is created by elastically deforming the artifact by attaching it to a similar flat plate with tapped holes using six equally spaced socket head cap screws in the counterbores near the measurement band. The six screws are tightened to yield which results in consistent deformation in the region of each screw. While the screws are still tight (in the deformed state), the measurement band is lapped flat better than 100 nm. After lapping, the screws are removed and the resulting form is shown in Figure 2.

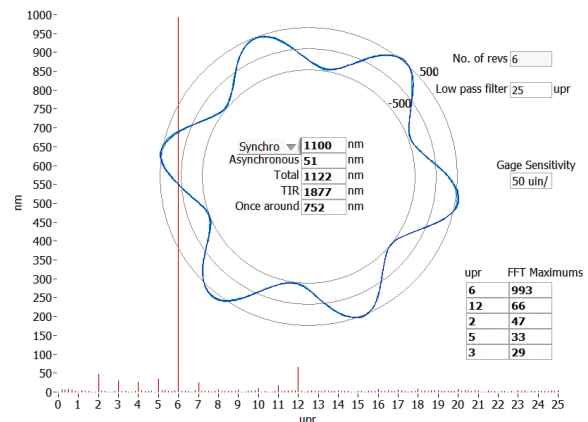


FIGURE 2. Flatness profile of the fidelity artifact showing 1 μm of six-lobe and 0.8 μm out-of-parallelism.

Since dynamic response is dependent on displacement amplitude, the micrometer-level form is ideal. Typical production parts require better than 0.25 μm amplitude fidelity. An artifact with 1 μm form provides a coverage factor of four times a typical production part.



FIGURE 3. Cartridge type LVDT with 250 mN contact force and 150 nm resolution.

The styli under test are shown in Figure 3, Figure 4, and Figure 5. The cartridge type plunger Linear Variable Differential Transformer (LVDT) shown in Figure 3 is a popular stylus choice due to its robust design. It is limited by the friction in the plain sleeve bearing but can be fitted with a diamond indicator tip. Diamond tips are preferred over ruby due to lower friction and wear. Typical resolution of this indicator is 150 nm.

The lever head LVDT shown in Figure 4 uses a pivot bearing with improved friction characteristics. This indicator provides greater flexibility and can provide better access to the gaging surface but diamond tips are not readily available. The resolution is better than 100 nm.



FIGURE 4. Lever head LVDT indicator with pivot bearing and 40 mN contact force and better than 100 nm resolution.

Finally, the ultimate in high resolution contact styli is the air bearing C-LVDT (Lion Precision) shown in Figure 5. The porous graphite air bearing virtually eliminates friction and hysteresis. It accepts a standard diamond tip and uses a capacitive sensor, rather than LVDT, to provide better than 10 nm resolution.



FIGURE 5. Air bearing C-LVDT (Lion Precision) with adjustable contact force and better than 10 nm resolution.

Use of the artifact with this variety of stylus pick-ups at scanning speeds typical of roundness testers can lead to significantly different results due to the variable frequency response of each stylus [1]. This is analogous to the trackability of gramophone pick-ups and their ability to stay in contact with the vinyl surface [2]. To characterize this dynamic response, the circular flatness of the artifact was measured with each stylus over a range of speeds.

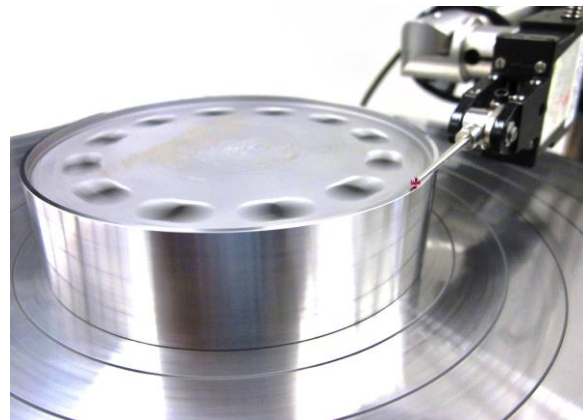


FIGURE 6. Characterizing the dynamic response of the lever head LVDT on a roundness tester.

The measurement speed was varied from 10 RPM to 120 RPM. With 1 UPR (undulation per revolution, aka, lobes) and 6 UPR spatial frequencies, this provides a frequency range from 0.17 Hz to 12 Hz and is consistent with expected roundness and flatness measurement conditions.

$$f = \frac{N}{60} \times L \quad (1)$$

where f is the frequency in Hz, N is the speed in RPM, and L is the spatial frequency in UPR.

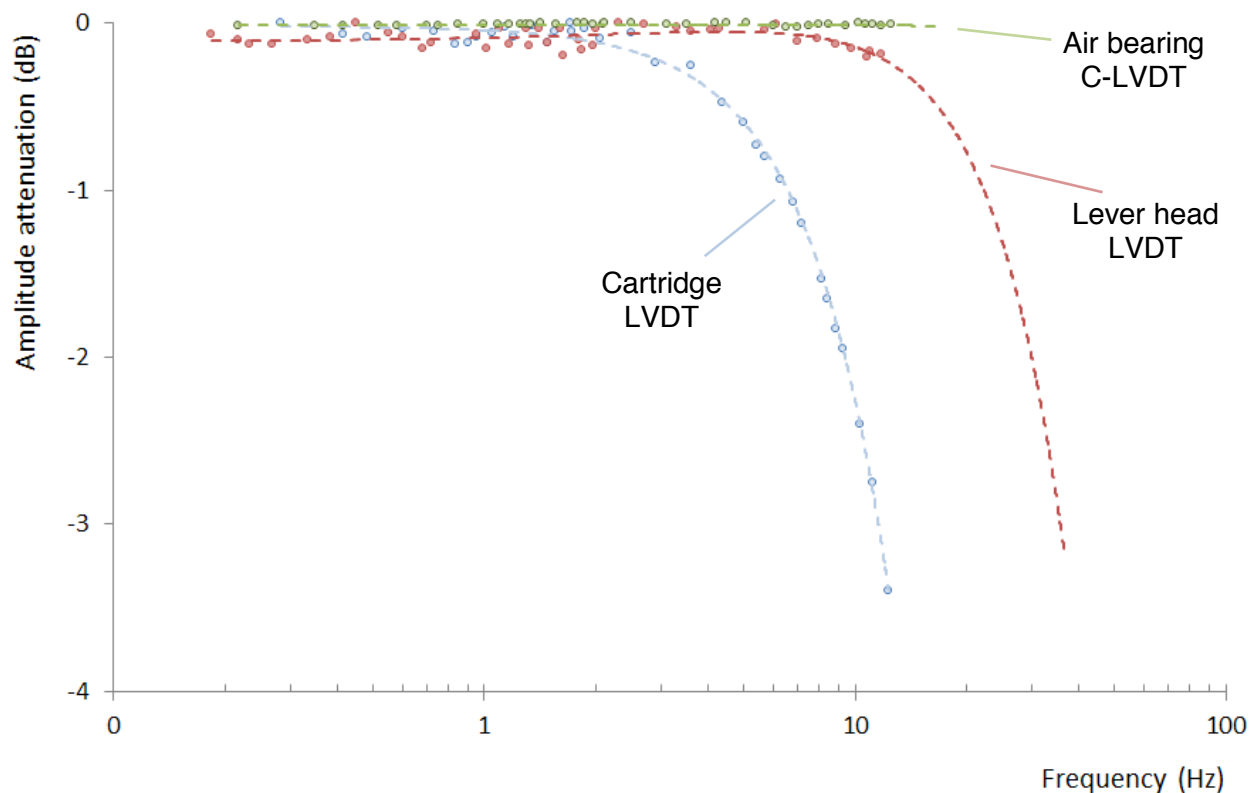


FIGURE 7. Amplitude attenuation as a function of frequency for three styli.
Reference amplitudes are $0.8\ \mu\text{m}$ at 1 UPR and $1.0\ \mu\text{m}$ at 6 UPR.

RESULTS

Amplitude attenuation as a function of frequency for three styli is shown in Figure 7. The measurement bandwidth for each stylus at -3 dB can be estimated using this information. The cartridge LVDT stylus starts showing trackability errors at about 1 Hz and by 10 Hz the amplitude is -3 dB from the reference. Similarly, the lever head LVDT begins rolling off at 7 Hz and its bandwidth is about 30 Hz. In contrast to these is the air bearing C-LVDT which shows no tracking errors over the frequency range.

CONCLUSION

An in-process metrology feedback loop for roundness and circular flatness is critical to the production of air bearing spindles. From over tightening the clamp on the cartridge-type LVDT, to loose indicator tips, to improper calibration adjustment, electronic indicators are notorious for giving erroneous results and require regular verification. The design and manufacture of an artifact for dynamic calibration is described. This test also has the advantage of testing the whole

structural loop of the instrument, not just the gage, and will identify problems such as loose joints and structural resonances.

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EFFECT OF TITANIUM DIOXIDE MIXTURE CONCENTRATION ON MATTE COATING AND 3D SCANNING ACCURACY

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INTRODUCTION

Accurate inspection measurements are necessary in many industrial applications. Optical 3D scanners (laser, fringe projection) are used more and more often, replacing CMMs because of their benefits in speed, number of acquired measurement points and also increasing accuracy.

Industrially produced parts are often problematic for structured light scanning because they are made of reflective or dark (thus, hard to scan) materials such as metals or some plastics or they are even translucent such as some plastics or glass. Parts made from such materials are difficult to scan or it can be even impossible to capture the surface. In such cases special temporary coatings are applied on the surface of the part to be scanned. These coatings usually make thin white matte layer that can be easily removed or evaporates on its own after some time.

Even though the thickness of the coating is small it still changes the dimensions of the measured part and negatively influences the accuracy of the inspection process. The thickness of the coating layer and its effect on the dimensions of the measured part was already evaluated in a study from Paloušek et al. [1]. This study was focused especially on measurement of chalk coating on precise calibrated elements. Other study in this field from Brajlíh et al. [2] mentions the problem of appropriate coating application and its influence on dimensional deviations. Errors of measurement caused by cyclododecane and chalk coatings were compared in [3].

Nowadays there is no international standard for measurement with optical scanners. Even though there are efforts to bring a standard to the field of close range systems there is no noticeable progress [4] and the only available and used document in this area is the guideline VDI/VDE 2634 and for the triangulation based systems especially its parts 2 and 3 [5] [6].

As optical scanners are becoming standard in inspection there is also a need to classify the

errors and effects caused by coatings applied to matte the surface of scanned part.

AIMS OF THE WORK

For application of the titanium coating theoretically any concentration of the TiO_2 and ethanol suspension can be prepared. The only limitation is the throughput of the airbrush gun. Based on this consideration a question arises. How and to what extent does the concentration of the suspension influence the results of scanning such surface?

The main aim of this work is to find the optimal concentration of the suspension. The optimal concentration will be selected based on experiments and such concentration should provide sufficient and good covering and the thinnest possible layer.

The next aim of this work is to prove or disprove the influence of the concentration on the layer thickness and also on the dimensions measured on the calibrated element. The next aim is to show how the coatings with different concentrations influence the scan quality of the flat surface expressed by the deviations from the ideal plane.

MEASUREMENT METHODOLOGY

The application of titanium coating was performed under the same conditions using an automatic travel for the airbrush gun (FIGURE 1). Conditions used for the titanium coating application:

- number of sprayings: 1
- speed of the gun: 0.224 m.s^{-1}
- spraying distance: 100 mm
- type of the airbrush gun: iwata HP-C Plus, nozzle diameter 0.3 mm
- air pressure: 1 bar
- opening of the gun: one revolution
- coating: Kronos TiO_2 + ethanol ES 200-578-6
- TiO_2 /ethanol weight ratios: 1:1 – 1:30

Based on preliminary results we applied also four sprayings of concentration 1:12 to compare it to one layer of concentration 1:3 and analyze the influence of application the coating layer in more sprayings.

We used three different samples (shown in FIGURE 1) for different experiments evaluations:

- silicon plate – layer thickness, profile of the layer and coverage ratio
- mirror polished stainless steel block – scanned surface coverage
- gauge block – changes in calibrated dimension

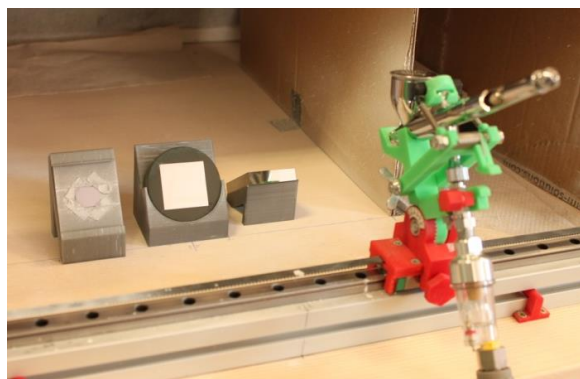


FIGURE 1 Airbrush gun on the travel and measurement samples

Methodology of 3D scanning

The capability of the scanner to capture the surface of the stainless steel block and dimensions of the gauge block were evaluated using ATOS III Triple Scan, 3D scanner from GOM mbH company. Measuring volume MV 170 was used. Scripts were made for the scanning of the samples in order to guarantee repeatability of the methodology. Positions of the samples during scanning, parameters of the scanning and process of the evaluation were stable during all experiments.

Distance was measured on the nominal dimension of the 50 mm gauge block in 5 points projected perpendicularly to the opposite plane. It was scanned from two orientations and both scans were transformed by 4 reference points.

Stainless steel block top surface was scanned in two orientations relative to the scanner, 0° and 45°, to evaluate the influence of the coated surface orientation on the capability to be scanned. Polygonal data were smoothed (0.005 mm) to eliminate the influence of measurement

noise and ideal plane was fitted to the area 20 x 20 mm.

Methodology of surface profile measurement

Layer thickness, profile of the layer and coverage ratio was measured on a silicon plate (13 x 15 mm) using Contour GT-X (Bruker, USA) profilometer. We used 50X magnification to capture the relief of the coating layer and 5X magnification to capture the grayscale image for coverage ratio evaluation. The images were evaluated by Vision64 software.

For the images with 5X magnification we used the Grayscale Data function to convert information about the light intensity to 256 grayscale image (see FIGURE 3). For each image (concentration) a histogram was made and based on the histogram for concentration 1:1 we determined the grayscale level 85 as the middle point between dark (covered) and light (uncovered) pixels. This fixed level was used for coverage ratio determination for all concentrations (results in FIGURE 2).

RESULTS

Feasibility of the coatings

During the experiments we found out that with concentration 1:1 of the TiO₂ and ethanol solution the airbrush nozzle starts to clog immediately after starting to spray. In the case of concentration 1:2 the airbrush started to clog after few seconds. These concentrations are too dense and unusable for practical application. Solutions with concentrations 1:3 – 1:30 are applicable without any problems and thus are suitable for practical application.

Coverage ratios and coating appearance

Observation of the silicon plate on profilometer proved the concentrations 1:1 and 1:2 as unsuitable. In the case of concentration 1:1 the coating has a form of sparsely distributed stains. The concentration 1:2 shows better results but it still doesn't reach the coverage of concentration 1:3. With percentage coverage 98.04% it proved to be optimal from the coverage point of view. Solutions with subsequent concentrations show linear decreasing trend in the coverage as was expected and as can be seen in graph in FIGURE 2.

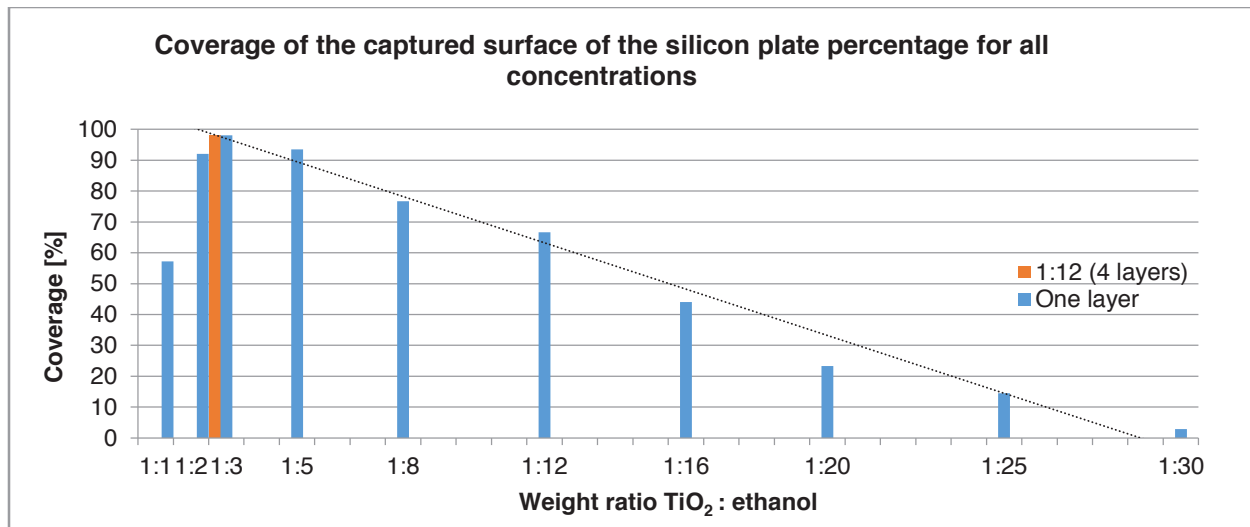


FIGURE 2 Coverage percentage of the silicon plate for all concentration

FIGURE 3 shows how with decreasing concentration and also with decreasing density of the matte coating the covering effect is decreasing and the intensity of reflected light is increasing. Concentration 1:30 shows very poor coverage of the surface.

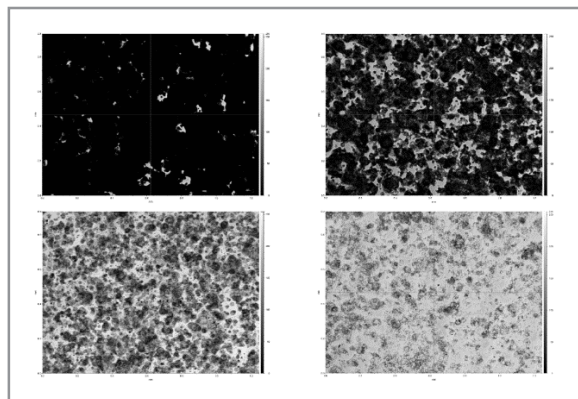


FIGURE 3 Grayscale images of the coating (concentrations 1:3, 1:8, 1:20 and 1:30).

Relief of the matte coating from the profilometer measurement was only observable for concentrations 1:1 – 1:3, images from other concentrations contain areas with wrongly evaluated values below zero that are caused by partial reflection of light. Relief of the coating with concentration 1:1 and two perpendicular sections of this relief is shown in FIGURE 4. Larger TiO_2 particles were observable also only for concentrations 1:1 – 1:3.

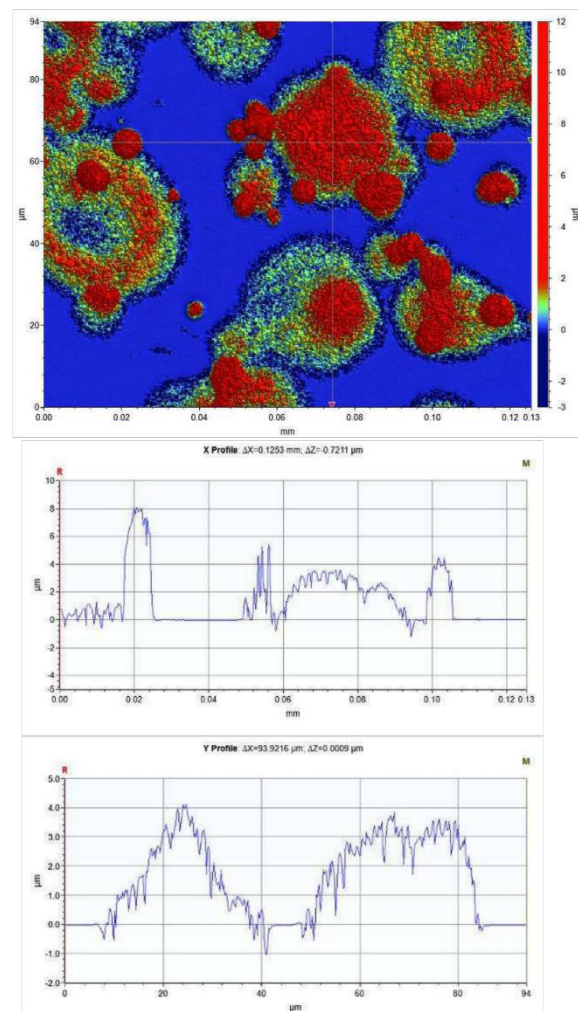


FIGURE 4 Relief of the coating layer (concentration 1:1) and two sections.

Sufficiency and quality of the coating

The quality of the coating was evaluated based on the deviations of the captured data of the polished stainless steel block top plane (see FIGURE 5) from the fitted plane. Data were captured for two angles of the stainless steel block relative to the sensor. Maximal deviations in both directions from the ideal plane for these two angles and all concentrations are presented in FIGURE 6 and FIGURE 7. These graphs show that the smallest deviations are reached with concentrations 1:2 – 1:5. From comparison of these graphs it is also evident that with 0° rotation of the sample the deviations are larger than with 45° rotation. More reflected light rays aim towards the camera in the first case and result in noisy or saturated pixels.

Coating with concentration 1:12 applied in four layers show similar deviations as concentration 1:3 applied in one layer. Quality of the coating seems to be independent on the number of layers in which specific amount of the TiO_2 is applied.

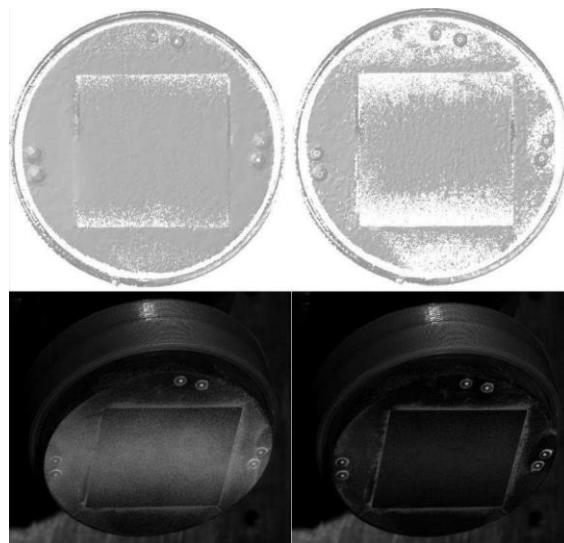


FIGURE 5 Coated surface (down) of the polished stainless steel block with concentrations 1:3 (left) and 1:30 (right) and resulting scans (up)

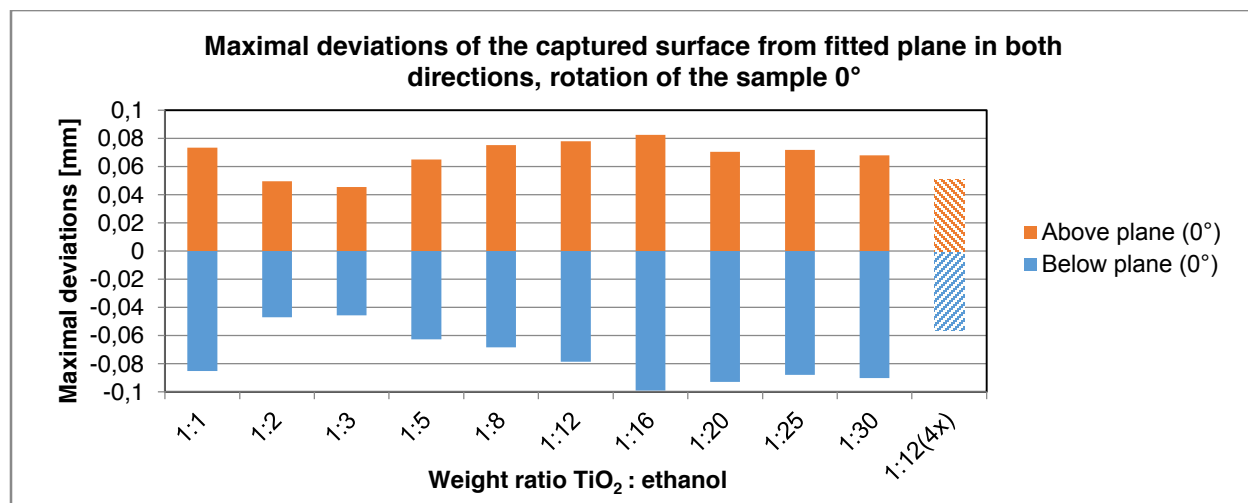


FIGURE 6 Maximal deviations of the captured surface from fitted plane (rotation of the sample 0°)

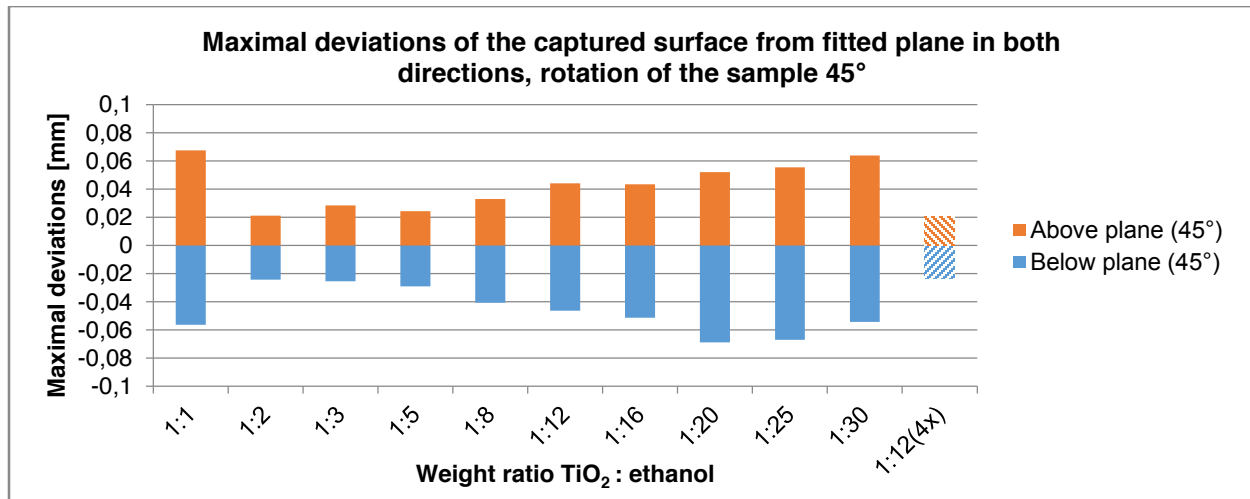


FIGURE 7 Maximal deviations of the captured surface from fitted plane (rotation of the sample 0°)

Thickness of the coating

It was already mentioned that the thickness of the coatings applied in our experiments could not be measured on profilometer. For evaluation of the layer thickness we used precise calibrated gauge block and we measured the changes in the calibrated dimension. Scanning of the gauge block showed that the expected decrease of the layer thickness with decreasing concentration of TiO_2 stops shortly after concentration 1:3. Differences for following concentrations are very small and on a similar level as the resolution capacity of the scanner, 2,5 – 6 μm . Layer thicknesses calculated from the distances

between five points and their projections to opposite plane on the gauge block are presented in FIGURE 8. Interesting is a noticeable decrease of the layer thickness for four layers of concentration 1:12. Part of the applied particles could have been blown off during subsequent applications of the coating. Second explanation could be that TiO_2 particles applied during subsequent sprayings were squeezed between already applied particles and resulted in continuous thinner layer. More experiments need to be performed to verify the influence of application of given amount of TiO_2 in multiple layers instead of at once.

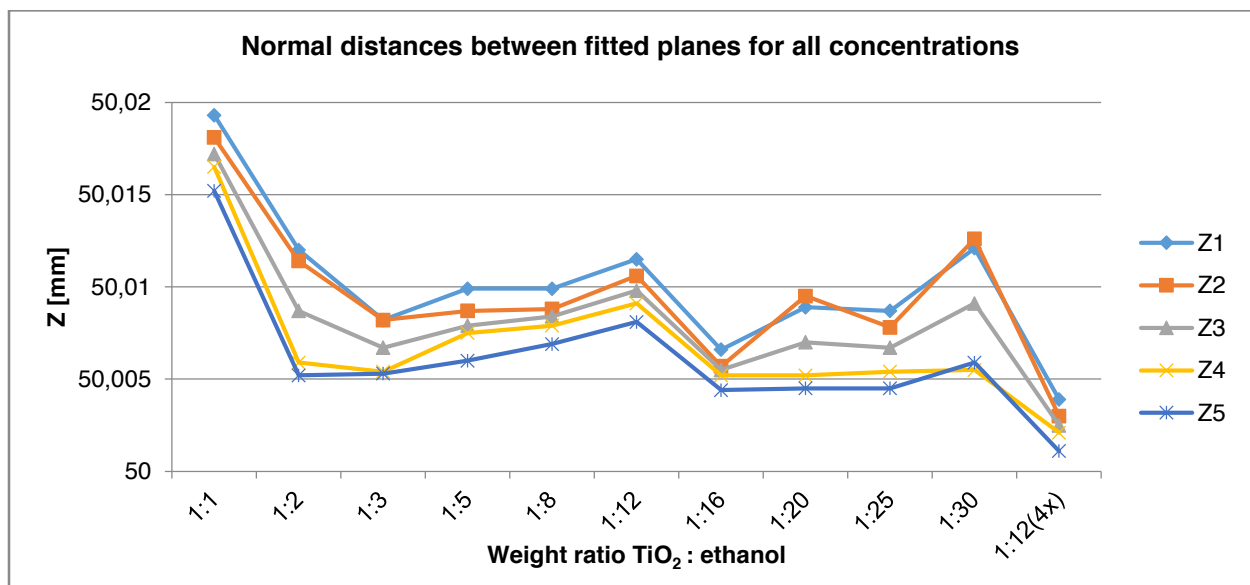


FIGURE 8 Normal distances between walls of the nominal dimension of the gauge block

CONCLUSION

In this study we have studied the influence of the TiO₂/ethanol solution concentration on the layer thickness and the ability of the 3D scanner to capture the surface. Solutions of TiO₂ and ethanol with concentrations larger than 1:3 are unusable because the solution is too dense and the airbrush gun starts to clog. The quality of the coating measured by deviations from the fitted plane is best for concentrations 1:2 – 1:5. This is because the surface is fully covered by the coating layer. In sparser concentrations the surface is not fully covered and partial light reflections result in these deviations. Scanning of the tilted surface brings results with smaller deviations of the scanned surface compared to the surface perpendicular to the scanner. Light rays are reflected out of the camera and do not result in noisy or saturated pixels. Results show that for one-layer coating a concentration in range 1:3 – 1:5 is best by the layer thickness and quality of the coated surface criterion. The effect of applying a given amount of TiO₂ in multiple layers instead of at once will be further investigated.

ACKNOWLEDGEMENTS

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Evaluation of Two Rotary Motion Stages using ISO 230-7 ANSI B89.3.4 Methods

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INTRODUCTION

Rotary motion stages are used as a rotational axis on traditional CNC machine tools, Indexing stations on complex machine tools and as a critical element of inspection instruments. The accuracy of rotation of the rotary stage limits the results in the parts produced and measurements made. There are accepted standards that specify how a rotary stage should be measured (ISO 230-7¹ and ANSI/ASME B89.3.4²). Depending on the intended use of the rotary stage, certain error motions in the standard are more or less important³.

The desired motion of a rotary stage (or any axis of rotation) is pure rotational motion. That means that any motion other than rotational motion is an error motion. Possible error motions include

- Axial (along the axis)
- Radial (perpendicular to the axis)
- Tilt (the extent that the error gets worse as the distance from the stage surface increases)

With an axis of rotation, there are 3 classifications of errors

- Synchronous – errors that repeat at the same angle on each rotation (repeatable)
- Asynchronous – errors that do not repeat at the same angle (non-repeatable)
- Drift – errors caused by temperature change or some operating condition

Measurement results are shown in 2 formats:

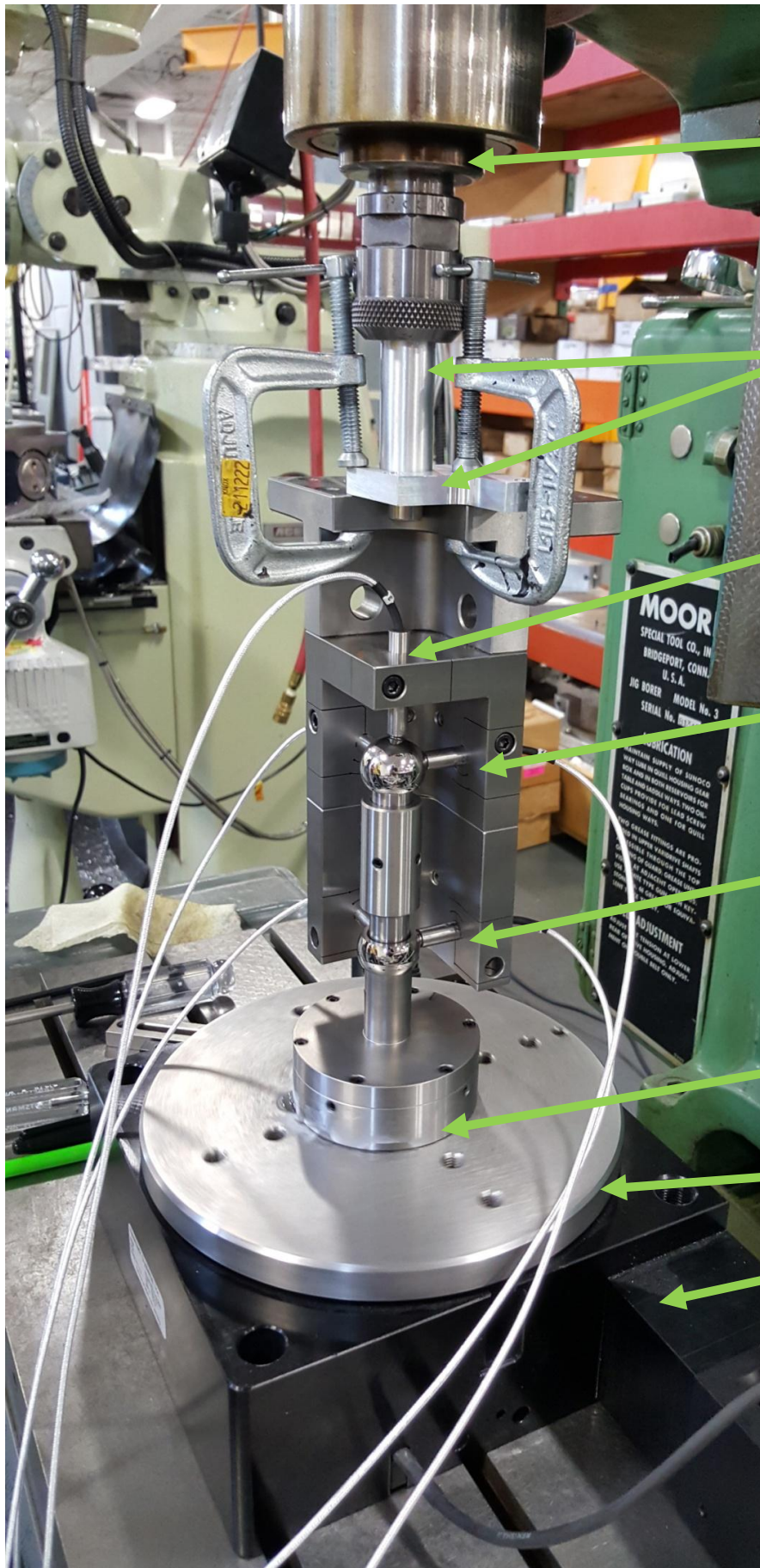
- Error Motion Value which is calculated from the measurement data
- Polar Plot – a graphical representation which can be helpful to assign cause.

SCOPE

Testing was completed in 1 day at the facilities of MTA (Motion Tech Automation). Rotary stages from 2 manufactures are compared and are referred to in this document as “Stage A” and “Stage B”. The results in this paper compare the Radial and Tilt Synchronous and Asynchronous errors. No thermal drift tests were performed

SETUP

The stages were driven by a servo motor with a 2048 line encoder. Stage A has an 80:1 gear reduction and Stage B has a 20:1 gear reduction. A Moore Special Tool boring machine was the platform for the measurements. A standard “Dual Ball Master” (S/N 658) from Lion Precision was used as the “Master Target”. Each sphere on the Dual Ball Master is certified to be round to 50 nanometers (0.050 micrometers or 2 millionths of an inch). The Dual Ball Master was set on the rotary stage, centered by tapping into place and then “super-glued” in place prior to taking measurements. The ball located closer to the rotary stage table is referred to as the “inner ball” and the ball located farther is referred to as the “outer ball”. “Phase A+” and “Phase A-” from the servo motor encoder were input into the Encoder input on the Lion Precision TMP190 Module. Five Lion Precision capacitive probes were mounted into a standard probe nest and then mounted in the spindle of the Moore Boring Machine using an adaptor. This is the preferred configuration according to the ISO230-7 Standard. See photo on next page.



Machine
Tool Spindle

Adaptor to 'hang' probe
nest from spindle

Capacitive Probe in
nest aligned with outer
ball in (Z direction)

Capacitive Probe in
nest aligned with
outer ball (Y direction)

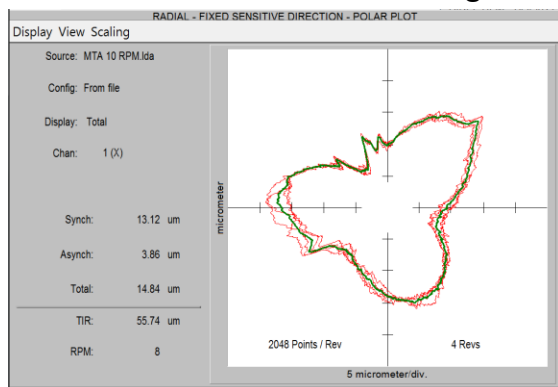
Capacitive Probe in
nest aligned with
inner ball (Y direction)

Master ball fixture
Super Glued to table

Rotary Stage:
Rotating Table

Rotary
Stage: Base

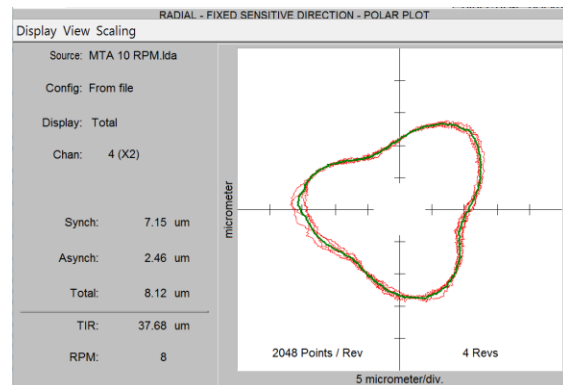
RADIAL MEASUREMENTS of Stage A



PLOT 1. Stage A Radial motion in the **X** direction on the outer ball over 4 revolutions

Explanation: This plot shows results at the outer ball as measured by the probe in “X position”.

1. The Synch (synchronous error) is 13.12 um. This is a measure of the “out of roundness of rotation”. The pattern is obviously triangular in shape. This suggests that there is something in the construction that constrains the bearings, shaft or housing in 3 locations. This is not a very good result. Rolling element spindles are Excellent if less than 1 um, acceptable if under 5 and anything higher is considered suspect.
2. The Asynch (asynchronous error) is 3.86 um. This is a measure of the repeatability of the spindle over multiple revolutions. For these 4 revolutions the worst case repeatability is 3.86 um. For a normal machine tool spindle, the analysis is done on a minimum of 20 revolutions. This value does not seem unusual.
3. Total Error of 14.84 um is the not the simple sum of the Synch and Asynch, but instead is based on evaluation of the polar plot of all revolutions instead of the synchronous/average plot which is shown in green.
4. TIR (total indicator reading or runout) value includes both the error motion AND the eccentricity (off center) of the target. The eccentricity is NOT an error motion. The analysis of the data eliminates the effect of eccentricity from the Synch and Asynch calculations. This why it is not possible to simply use an indicator or sensor to measure spindle error motions.

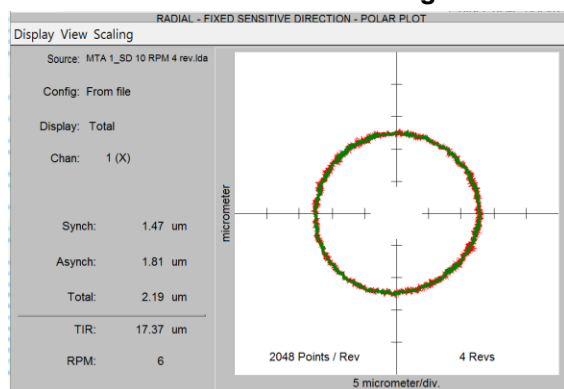


PLOT 2. Stage A Radial motion in the **X2** direction (inner ball) over 4 revolutions

Explanation: This plot shows results at the inner ball as measured by the probe in “X2 position”.

1. The Synch (synchronous error) is 7.15 um. The value is almost 2x better than the measurement at the outer ball. This is expected, as the errors become smaller as the measurement is taken closer to the face of the rotary stage. The pattern is still obviously triangular in shape and has the same orientation as the outer ball. This is not a very good result. Rolling element spindles are excellent if less than 1 um, acceptable if under 5 and anything higher is considered suspect.
2. The Asynch (asynchronous error) is 2.46 um. This value is also lower than the measurement at the outer ball, as expected.
3. Total Error of 8.12 um is also about half as the total error at the outer ball.
4. TIR (total indicator reading or runout) is also less at the inner ball: 37.68 um. This is inconsequential because the eccentricity is separated from the error motion as part of the analysis.

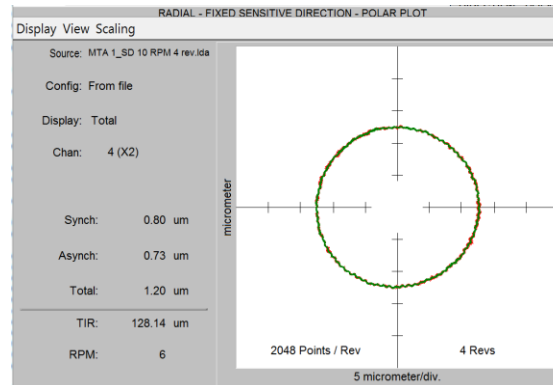
RADIAL MEASUREMENTS of Stage B



PLOT 3. Stage B Radial motion in the X direction (outer ball) over 4 revolutions

Explanation: This plot shows results at the outer ball as measured by the probe in the "X position".

1. The Synch (synchronous error) is 1.47 um. This is almost 10 times better than "Stage A" At this scaling, the plot is quite round. This is a reasonable result for a rotary table. To repeat the above comment: Rolling element spindles are excellent if less than 1 um, acceptable if under 5 and anything higher is considered suspect.
2. The Asynch (asynchronous error) is 1.81 um. This is a measure of the repeatability of the spindle over multiple revolutions. This value is about half of the value of "Stage A".
3. As explained above, the Total Error of 2.19 um is the not the simple sum of the Synch and Asynch.
4. TIR (total indicator reading or runout) value includes both the error motion AND the eccentricity (off center) of the target. The eccentricity of 17.37 um is NOT an error motion. The analysis of the data eliminates the eccentricity from the Synch and Asynch calculations. This why it is not possible to simply use an indicator or sensor to measure spindle error motions.



Plot 4. Stage B Radial motion in the X2 direction (inner ball) over 4 revolutions

Explanation: This plot shows results at the inner ball as measured by the probe "X2 position".

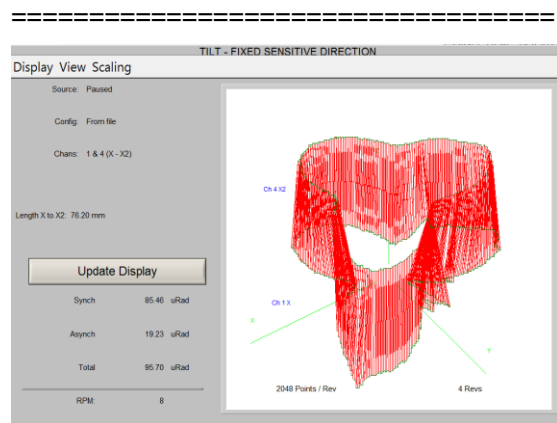
1. The Synch (synchronous error) is 0.8 um. The value is almost 2x better than the measurement at the outer ball. This is expected, as the errors become smaller as the measurement is taken closer to the face of the rotary stage. The pattern is very round. At this distance from the stage table, the results are excellent. Rolling element spindles are excellent if less than 1 um, acceptable if under 5 and anything higher is considered suspect.
2. The Asynch (asynchronous error) is 0.73 um, about 30% of Stage A and about half of the "X position" measurement.
3. Total Error of 1.20 um is also about half as the total error at the outer ball.
4. TIR (total indicator reading or runout) is higher at the inner ball: 128.14 um. This is inconsequential because the eccentricity is separated from the error motion as part of the analysis.

TILT MEASUREMENTS OF STAGES A & B

Another critical characteristic of a rotary stage is the tilt error. As observed above, if the point of interest is farther from the surface of the table, the result degrades. This degradation is quantified as the “Tilt Error”. The users of rotary tables want the height as low as practical, which puts the bearings close together, which makes it very difficult to maintain tilt performance. One of the simplest ways to have better tilt performance is to move the spindle bearings apart (i.e. have a longer spindle shaft). This basic constraint of what the customer wants (minimal height) and the optimum mechanical configuration (maximum height) represents a classic design tradeoff.

To measure the tilt, measurements are taken at both the outer and inner ball simultaneously. These 2 sets of data are then analyzed to identify the tilt error. Data is presented in a 3-D plot.

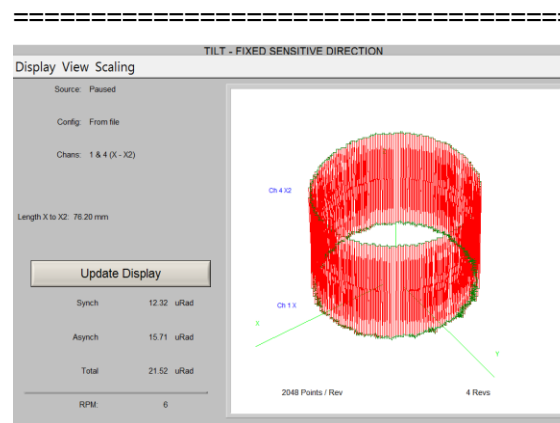
The reported value of tilt error is in “micro-radians”, which at first seems like a very strange unit of measure. For small angles, the micro-radian can be expressed as micrometer per meter (or micro-inch per inch). That is to say if a measurement result is 20 micro-radians, the error will grow at a rate of 20 micrometers per meter (or 20 micro-inches per inch which is the same as 1 ‘tenth’ per 5 inches).



PLOT 5. Stage A Tilt motion in the **X** direction over 4 revolutions.

Explanation: This plot shows tilt results in the X direction as measured by the X and X2 probes

1. The Synch (synchronous error) is 85.46 micro-radians. This is a measure of the how the “out of roundness of rotation” degrades as the distance from the rotary table increases. The pattern is obviously 3 lobe.
2. The Asynch (asynchronous error) is 19.23 micro-radians.
3. In this test result, it is obvious that Total Error of 95.70 micro-radians is the not the simple sum of the Synch and Asynch values.



PLOT 6. Stage B Tilt motion in the **X** direction over 4 revolutions.

Explanation: This plot shows tilt results in the X direction as measured by the X and X2 probes.

1. The Synch (synchronous error) is 12.32 micro-radians. This is a measure of the how the “out of roundness of rotation” degrades as the distance from the rotary table increases.
2. The Asynch (asynchronous error) is 15.71 micro-radians. This is a measure of the non-repeatability of the spindle over multiple revolutions
3. In this test result, it is obvious that Total Error of 21.52 micro-radians is the not the simple sum of the Synch and Asynch.

SUMMARY OF RESULTS:

Operating speed 6 to 10 rpm

Number of revolutions: 4

Measurements taken:

- Radial error motion in the X direction at outer ball: Synchronous, Asynchronous, Total Errors
- Radial error motion in the X direction at inner ball: Synchronous, Asynchronous, Total Errors
- Radial Tilt error motion in the X direction: Synchronous Asynchronous, Total Errors

	Radial Synch Outer (um)	Radial Synch Inner (um)	Radial Asynch Outer (um)	Radial Asynch Inner (um)
Stage A	13.12	7.15	3.86	2.46
Stage B	1.47	0.80	1.81	0.73
Improvement	9x	9x	2x	3x

Table 1. Summary of radial measurements

	Tilt Synch (u-rad)	Tilt Asynch (u-rad)
Stage A	85.46	19.23
Stage B	12.32	15.71
Improvement	7x	1.2x

Table 2. Summary of Tilt Measurements

CONCLUSION: Using tests established by ISO and ANSI standards, we can conclude with confidence that Stage B is significantly more accurate than Stage A.

ADDITIONAL WORK: Sensitivity of the application to each error source should be analyzed. Ability to “map” or correct for the various error sources should be analyzed. Techniques to improve repeatability based on the application should be considered.

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- [1] ISO 230-7:2015 Test code for machine tools- Part 7: Geometric accuracy of axes of rotation
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OSCILLATING BALL-BEARING-SPINDLE ERROR MOTION

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INTRODUCTION

There are indications in the literature that ball bearing spindle error motion is quite repeatable when the spindle is wound back to the original angle, then rotated forward again [1,2,3]. Marsh [1] provides measured radial error motion over three revolutions for a ball bearing that was rewound to the start position after 32 revolutions. The single ball bearing was held in a “bearing error motion instrument.”

The purpose of this report is to present some experimental error motion data on a more typical ball bearing spindle that is subjected to oscillating motion. In this case oscillating motion simply implies a general back-and-forth motion, not a round-and-round monotonic motion. Examples of devices that might have oscillating spindle motion include: robots, hard drive lever arm, cyclotron magnet mapper, CMM measurement head, survey equipment, roundness tester, machine tool rotary axis, scanner rotation stage, etc. In such cases it could be possible to reduce system errors by measuring and compensating for the repeatable spindle error motions.

Although hydrostatic bearings are considered the gold standard for accurate spindle motion, they are not always convenient or possible to use in which cases rolling element bearings are often chosen. A typical ball bearing might have asynchronous error motion of about 300 nm when rotating continuously, however due to the repeatability mentioned above the effective asynchronous motion is expected to be much less when oscillating.

Test results discussed include spindle error motion for: low cycle oscillating motion where one direction only is important, effects of temperature change, tilt error motion, error separation for artifact and spindle, oscillating motion where both directions are important, fundamental radial error motion, and higher numbers of cycles. Additional results are given to support these main topics.

EXPERIMENTAL SETUP

The spindle used for these experiments had a vertical axis with four SKF S7005 CDGB/P4A sealed angular contact bearings in the QBC (stiff) arrangement. The bearings had a light interference fit with the steel shaft and a 17-4PH stainless steel housing. Integral to the housing was an XY flexure stage for centering a precision artifact. The spindle assembly was mounted to a heavy ground steel plate, all resting on a granite flat, see Figure 1.

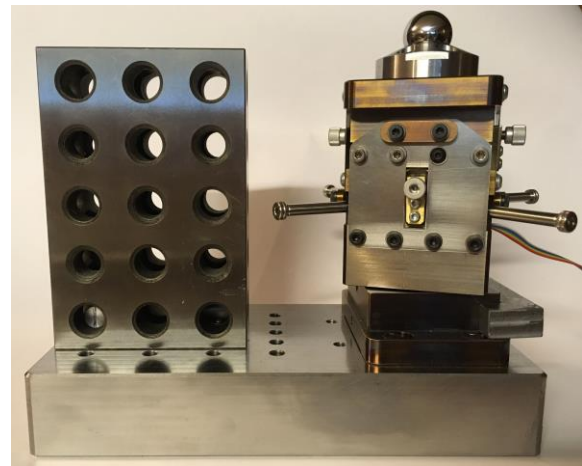


FIGURE 1. Experimental Setup without Probe and Holder.

The artifact was a 1" diameter GR2.5 stainless steel ball from Bal-tec part number: UB-B100 BL. The ball was glued to a custom holder that was bolted to the spindle face. Additional features were added to the spindle face and to the custom holder to facilitate accurate registration of the artifact at 180° angles. The artifact was located 104 mm above the middle of the bearing stack (low state). For some tests a spacer was used to increase the distance from the bearings to the artifact by 50 mm (high state). The artifact was calibrated at the National Research Council of Canada and found to have a measured roundness of 49 nm with an expanded uncertainty ($k=2.37$) of 22 nm.

Fixed sensitive direction radial motion was measured at the equator of the artifact with a Lion Precision capacitive probe model C8-2.0 and a CPL350 driver. The probe was mounted to a heavy steel beam at one of three locations spaced evenly at 90°. The probe signal range was adjusted to increase sensitivity with an API DC-to-DC transmitter part number: APD4380. Data acquisition was with a US Digital USB4-D device connected to a PC.

The data acquisition system was setup to record encoder and probe data 16,384 times per revolution. The spindle was not motorized and was rotated by hand. The encoder cable rotated with the spindle during the tests and care was taken to minimize forces applied to the spindle through the cable.

TERMINOLOGY

The vast majority of spindle error motion literature is concerned with monotonic rotation having a constant angular velocity. This is the type of motion discussed in the ASME Standard [2] where a detailed list of definitions and specifications are given. The terminology used in this report to discuss oscillating spindle motion relies heavily on the ASME Standard, but also requires some additional definitions:

Oscillating spindle runout: runout (radial, axial, etc.) over a specified angular range for an oscillating spindle. May include components that are not considered error motion such as artifact eccentricity for radial motion.

Oscillating spindle error motion: spindle error motion over a specified angular range for an oscillating spindle.

Oscillating spindle synchronous error motion: synchronous component of oscillating spindle error motion.

Oscillating spindle asynchronous error motion: asynchronous component of oscillating spindle error motion.

One-directional error motion: oscillating spindle error motion where only one direction (CW or CCW) is important

Bi-directional error motion: oscillating spindle error motion where both directions (CW and CCW) are important

Fundamental radial error motion: for both oscillating and monotonic spindle motion, the sinusoidal component of the radial error motion that occurs at one cycle per complete revolution (1 UPR) and is not related to artifact eccentricity.

In the rest of this report the word oscillating is frequently omitted and it should be understood from the context when it is implied. Oscillating generally means moving back-and-forth in a continuous harmonic type of motion, but a more general definition of any back-and-forth motion is used here.

DATA ANALYSIS

The 16,384 data points generated for each one-directional cycle were imported into a spreadsheet then averaged. Averaging reduced the size of the vector to 1024 by calculating the mean for each group of 16 points. Averaged measured results referred to in this report have been averaged in this manner. This averaging was necessary in order to use the spreadsheet FFT function which had a 1024 size limit.

For each 1024 vector of averaged measured data the FFT was calculated to convert to the frequency domain. The 0 UPR and 1 UPR components were often removed, then an inverse FFT was calculated to convert back to the spatial domain. Synchronous error motion was calculated by averaging the inverse FFT vectors. Asynchronous error motion was calculated by subtracting the inverse FFT vectors from the synchronous error vector. The standard deviation of the residuals was calculated by taking the standard deviation of the asynchronous error vectors.

TEST RESULTS

Ambient temperature stayed within $20.3 \pm 1^\circ\text{C}$ except for Test [624-2] where the lab was intentionally heated then cooled. Artifact angle with respect to the spindle and the probe orientation angle were not always maintained between tests. Other than as noted for Test [617-1/2], results include the effects of both spindle error motion and artifact form.

One-Directional Radial Motion [617-1]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for the low state in the CW direction only. The averaged measured radial runout for 10 cycles is shown in Figure 2. The 0 UPR component has been removed but the 1 UPR has not been removed from the measured data. Figure 3 shows the radial error motion with all 1UPR content removed. Figure 4 shows the synchronous and asynchronous components

with all 1 UPR content removed. The standard deviation of the residuals is 7.2 nm.

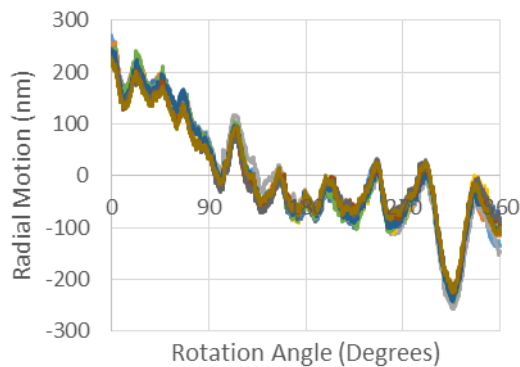


FIGURE 2. One-Directional Radial Runout from Ten Cycles.

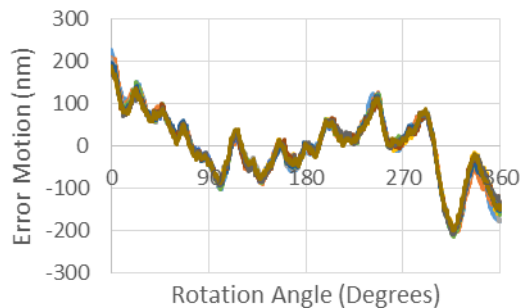


FIGURE 3. One-Directional Radial Error Motion from Ten Cycles.

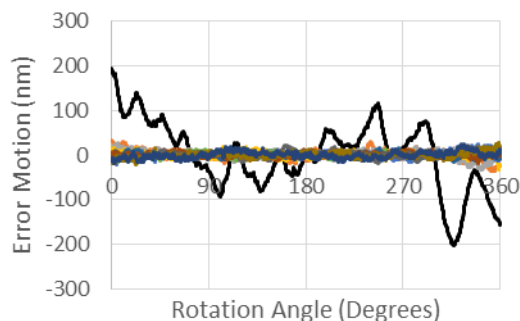


FIGURE 4. One-Directional Synchronous and Asynchronous Radial Error Motion from Ten Cycles.

The synchronous error motion at the start and end of the cycle are not the same, and this is expected for a rolling element bearing. The average amplitude of the 1 UPR content was about 100 nm, suggesting that the artifact eccentricity was no more than this amount. The

synchronous error motion is much larger than the asynchronous, opposite to what is typically observed for a monotonically rotating ball bearing spindle.

One-Directional Radial Motion: ΔT [624-1]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for the high state in the CW direction only. The temperature varied between 20.6°C and 24.1°C over about 1.5 hours. The averaged measured radial runout for 10 cycles is shown in Figure 5. A same constant value has been subtracted from the data for each cycle so that the average is close to zero, but the 1 UPR component has not been removed. Notice the vertical shift between the individual cycles due to the temperature change. Also notice the strong 1UPR content which indicates that the artifact was not centered very well. Figure 6 shows the radial error motion with all 0 UPR and 1UPR content removed. The standard deviation of the residuals is 6.4 nm.

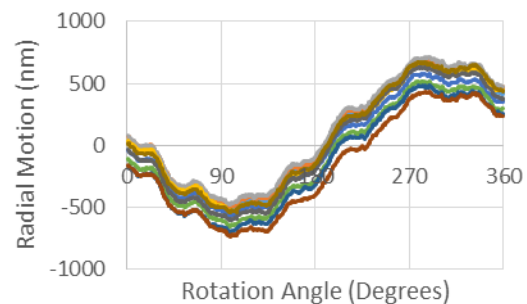


FIGURE 5. One-Directional Radial Runout from Ten Cycles with Temperature Change.

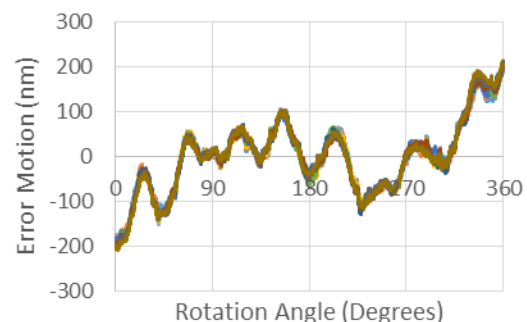


FIGURE 6. One-Directional Radial Error Motion from Ten Cycles with Temperature Change.

One-Directional Tilt Motion [623-1]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for both the low and high states in the CW direction only. The synchronous radial error motion with all 0 UPR and 1 UPR components removed for the low and high states are shown in Figure 7. Also shown in the figure is the calculated synchronous radial error motion that would be expected with the artifact 44 mm above the middle of the bearing stack, approximately where the synchronous radial error motion is smallest. Note that the calculation of this error motion did not account for the artifact form error, and is therefore approximate. Figure 8 shows the calculated synchronous tilt error motion. The synchronous tilt error motion was calculated as $(e_{sH} - e_{sL})/h$ where: e_s is synchronous error motion, subscripts H and L are for the high and low states, and $h=50$ mm.

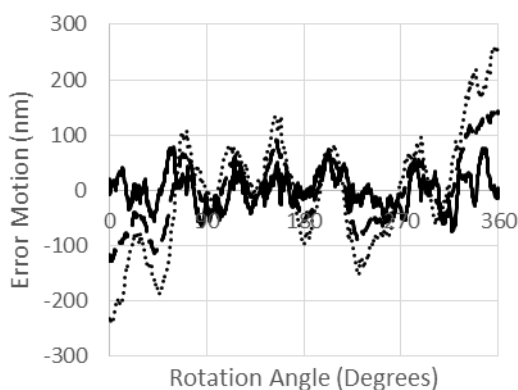


FIGURE 7. One-Directional Synchronous Radial Error Motion in low state [dashes], high state [dots], and calculated for 44 mm above the middle of the bearing stack (60 mm below the low state) [solid].

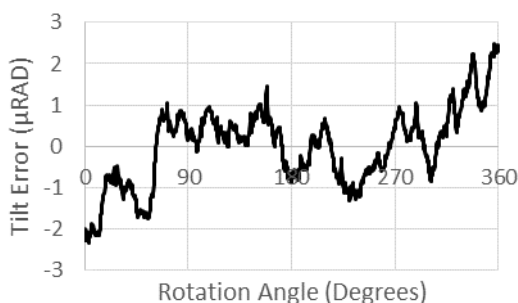


FIGURE 8. Calculated One-Directional Synchronous Tilt Error Motion.

Spindle & Artifact Error Separation [617-1/2]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for the low state in the CW direction only. The tests were conducted with artifact and sensor at two opposite orientation angles. The synchronous radial error motion for opposite orientation angles is shown in Figure 9. All 0 UPR and 1 UPR components have been removed. Donaldson reversal [3] was used to calculate the results for the spindle synchronous radial error motion with artifact form removed, and for the artifact profile as shown in Figure 10.

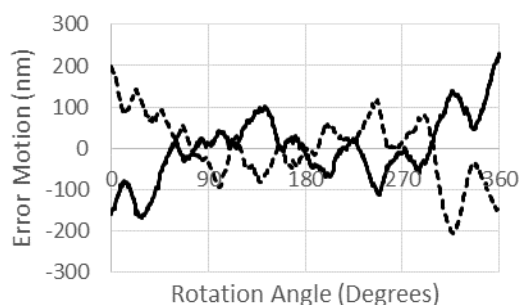


FIGURE 9. One-Directional Synchronous Radial Error Motion from Opposite Orientation Angles.

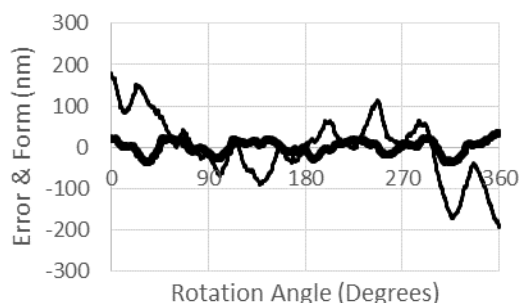


FIGURE 10. Calculated One-Directional Synchronous Radial Error Motion with Artifact Profile Removed [thin line], and Artifact Profile [heavy line].

The artifact LSC roundness [4] corresponding to the calculated profile is 74 nm. For the calculated results of Figure 10 the expanded uncertainty ($k=2$) was estimated [2] to be 56 nm. Recall that the artifact is specified as Grade 2.5 (maximum 63.5 nm deviation from spherical form), and has a roundness of 49 nm with an expanded uncertainty ($k=2.37$) of 22 nm as calibrated by NRC. The calculated results for

the artifact are consistent with the specification and with the calibrated roundness.



FIGURE 11. Polar Plot of Calculated Artifact Roundness Profile. 1 DIV = 50 nm.

Bi-Directional Radial Motion [620/21-1]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for the low state in the CW and CCW directions, and at two orientation angles spaced 90° apart. Figure 12 shows the synchronous radial error motion from one orientation angle with all 0 UPR and 1 UPR components removed. The standard deviation of the residuals of all data is 6.3 nm.

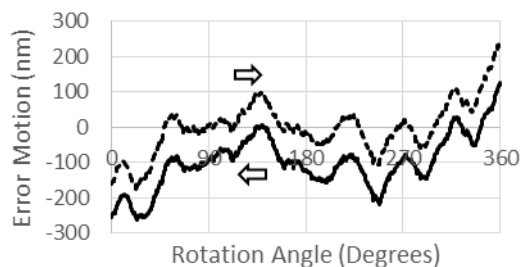


FIGURE 12. Bi-Directional Synchronous Radial Error Motion from One Orientation Angle in CW [dashed line] and CCW [solid line] directions. 100nm has been subtracted from the CCW data.

The two-dimensional measurement data was further analyzed by the method outlined in [5], similar to that in [6] and [7]. Two items of note from the analysis are that an axis shift of about 270 nm occurred between CW and CCW rotation, and that the fundamental radial error motion did not significantly change between CW and CCW rotation. In addition, the magnitude of the fundamental radial error motion was 60 nm (Rc/2 from [5]).

Bi-Directional Transient Motion [616-T5]

For these tests the spindle was rotated back-and-forth four times through one revolution and radial motion was measured for the low state in the CW direction. Upon rewinding each cycle the spindle was not returned as far as for the previous cycle. Figure 13 shows the measured motion over 180°. Note that there is a transient zone where the motion for subsequent cycles approaches the previous cycles.

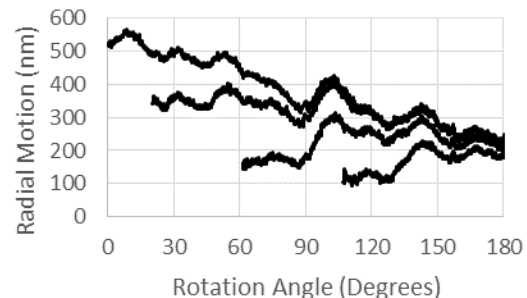


FIGURE 13. Bi-Directional Radial Motion Transient Zone.

One-Directional Motion: 100 Cycles [624-2]

For these tests the spindle was rotated back-and-forth through two revolutions, and radial motion over the middle 360° was measured for the low state in the CW direction only. The radial error motion for 10 of 100 cycles is shown in Figure 14 where all 0 UPR and 1 UPR components have been removed. The standard deviation of the residuals is 14.1 nm.

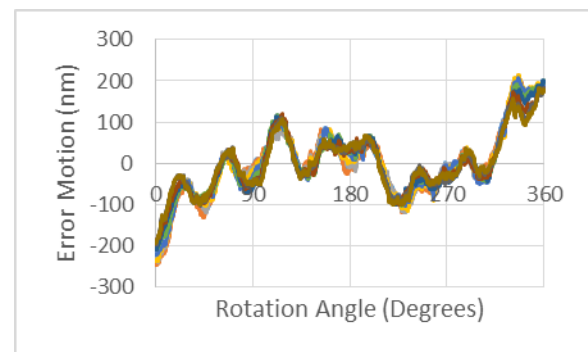


FIGURE 14. One-Directional Radial Error Motion over 100 Cycles (10 cycles shown spaced over 100).

Additional Test Results

In addition to the above, several other tests were conducted to provide a bit more information about the setup.

A capped test of the sensor showed that the noise floor of averaged data over 10 seconds had a peak-to-peak value of 2.8 nm. The same test with the sensor measuring the stationary test setup had a value of 3.5 nm.

Static Stiffness of 22 N/ μ m was measured with displacement at the artifact in the low state and the force applied 17 mm above the middle of the bearing stack. This is the same position where force was applied to rotate the spindle.

The force applied to rotate the spindle was measured to be about 0.35 N. The corresponding displacement of the artifact based on the stiffness measurement is expected to be about 16 nm. Some effort was taken to maintain a consistent rotation force between tests, however a motorized spindle would be better.

Total hysteresis of 3 nm for the non-rotating spindle and XY stage assembly was measured by displacing the artifact radially in opposite directions by about $\pm 0.15 \mu$ m. Hysteresis of the rotating spindle was not measured.

Thermal drift was measured in the high state and the probe to artifact distance changed by about 100 nm/ $^{\circ}$ C.

Speed of rotation for the ten cycles of Test [617-1] was calculated to be between 1 and 3 RAD/s. Neither acceleration nor jerk were calculated. Additional tests were conducted where the spindle was stopped then restarted mid-cycle without a noticeable difference in the spindle error motion. At higher speeds inertial effects will probably be significant. At very high speeds contact might be lost between the balls and the rings due to formation of a hydrodynamic film which could drastically reduce repeatability of the spindle error motion.

CONCLUSIONS

Experimental results were presented for several instances of oscillating ball-bearing-spindle error motion. One-directional oscillating spindle synchronous radial error motion was repeatable over 10 cycles, less repeatable over 100 cycles, and not sensitive to small temperature changes. Bi-directional oscillating spindle synchronous radial error motion (including fundamental radial error motion) was also repeatable, but had a significant axis shift between CW and CCW rotation, and had a transient zone when rotation direction was reversed. The roundness of a

spherical artifact was measured using the axis of rotation of an oscillating ball bearing spindle, and was found to be consistent with the specification and calibration data. Rotation speed did not have a significant effect on spindle synchronous radial error motion over the range of slow speeds that were tested.

ACKNOWLEDGEMENTS

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SURFACE SHAPE MEASUREMENT OF LARGE CURVED SURFACE APPLYING IMPROVED SEQUENTIAL THREE-POINT METHOD

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BACKGROUND

Demands are increasing for highly-precise surface shape measurement of large curved surface such as lens and reflecting mirror for telescope. However, there is no useful method for the purpose because of inevitable difficulties in preparing large tangible datum and to avoid the influence of error motions in measuring operation.

On the other hand, the sequential three-point method, which does not require the tangible datum, is well known as a useful technique for on-machine straightness measurement of large parts [1]. However, if a zero position error between sensors exists, a large quadratic error component is superimposed on measured shape profile. When the specimen surface has large unevenness, the distance between surface and displacement sensors exceeds the working range of the sensors during measuring operation. In this paper, a design concept of a new surface shape measuring method of large curved surfaces is proposed using a sequential three-point method improved the disadvantages described above.

PRINCIPLE OF SHAPE MEASUREMENT BY SEQUENTIAL THREE-POINT METHOD [1][2]

Figure 1 shows the principle of sequential three-point method. The rigid sensor head, on which

three displacement sensors F (front), C (center) and R (rear) are fixed at equal spaces of d , is scanned along a straight line (X-axis) on the specimen surface to be measured. The surface shape profile of the specimen is numerically calculated using the discontinuous displacement data obtained at every space of d as described in the following. The surface shape profile of the specimen and the output signals of the front, center and rear sensors are expressed by $g(x)$ and S_{3F} , S_{3C} , and S_{3R} , respectively. Supposing that the translational and rotational error motions of the sensor head are e_y and e_θ , respectively, the output signals of the sensors are expressed by Eqs. (1) to (3).

$$S_{3C}(x_i) = g(x_i) + e_y(x_i) \quad (1)$$

$$S_{3F}(x_i) = g(x_{i+1}) + e_y(x_i) + de_\theta(x_i) \quad (2)$$

$$S_{3R}(x_i) = g(x_{i-1}) + e_y(x_i) - de_\theta(x_i) \quad (3)$$

To eliminate the error motions e_y and e_θ , the second differential output signals is calculated as expressed by Eq. (4), where ΔS_{3F} and ΔS_{3R} are the differences between output signals of the front and center sensors and between the center and rear sensors, respectively. Eq. (4) can be rewritten as the recurrence formula Eq. (5). The formula means that the shape profile at x_{i+1} is sequentially obtained from the profiles obtained

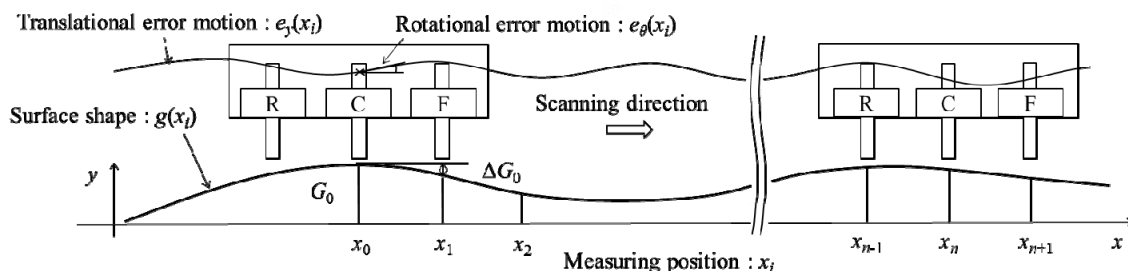


Figure.1 Principle of sequential three-point method

at x_i and x_{i-1} .

$$\Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) = \{g(x_{i+1}) - g(x_i)\} - \{g(x_i) - g(x_{i-1})\} \quad (4)$$

$$g(x_{i+1}) = 2g(x_i) - g(x_{i-1}) + \Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) \quad (5)$$

By replacing $g(x_0)$ with G_0 and $g(x_1) - g(x_0)$ with ΔG_0 , surface shape profile $g(x_n)$ without any error motions is expressed by Eq. 6. Where G_0 is the height of specimen surface at starting point x_0 of measurement and ΔG_0 is the difference height between x_0 and x_1 .

$$g(x_n) = G_0 + n\Delta G_0 + \sum_{i=1}^n (n-i) \{ \Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) \} \quad (6)$$

The error motions of sensor head e_y and e_θ can also be calculated from Eqs. (1) to (3) and (5). When the three sensors have different zero positions as shown in Figure 2, the zero position error α cannot be eliminated. A large quadratic error component $n(n-1)\alpha/2$ is superimposed on shape profile as shown in Eq. (7).

$$g(x_n) = G_0 + n\Delta G_0 + \sum_{i=1}^n (n-i) \{ \Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) \} + \frac{n(n-1)}{2} \alpha \quad (7)$$

SHAPE MEASUREMENT OF CURVED SURFACE BY IMPROVED SEQUENTIAL THREE-POINT METHOD [2]

When the measuring surface is almost flat with few undulations, the three-point method is a useful technique for on-machine measurement of its straightness because of simple and easy calculation algorithm. However, if the measuring surface has large irregularities such as convex or concave mirror, the distance between surface and sensor exceeds the working range of the sensor during measuring operation. Therefore, in improved three-point method proposed, distance between the measuring surface and the tip of center sensor is adjusted by a translational displacement h_a and inclination of the sensor head is adjusted by a rotational angle ϕ so that the measuring surface always comes within the working range of all displacement sensors as shown in Figure 3. The center of rotation is set to the tip of center sensor. The output signals of the three sensors are expressed by Eqs. (8) to (10).

$$S_{3C}(x_i) = g(x_i) + e_y(x_i) - h_a(x_i) \quad (8)$$

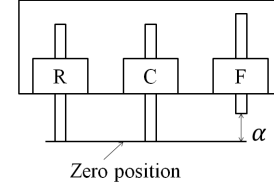


Figure.2 Zero position error between

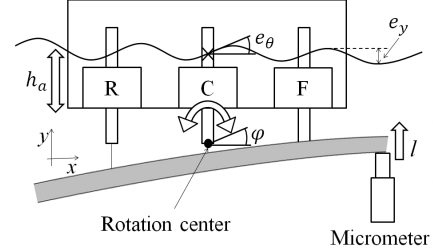


Figure.3 Principle of shape measurement of curved surface using improved three-point method

$$S_{3F}(x_i) = g(x_{i+1}) + e_y(x_i) + de_\theta(x_i) - h_a(x_i) - d \sin \phi(x_i) \quad (9)$$

$$S_{3R}(x_i) = g(x_{i-1}) + e_y(x_i) - de_\theta(x_i) - h_a(x_i) + d \sin \phi(x_i) \quad (10)$$

Eliminating e_y and e_θ as same manner as that in case without h_a and ϕ , Eq. (11) is obtained using the second differential output signals between ΔS_{3F} and ΔS_{3R} .

$$g(x_{i+1}) = 2g(x_i) - g(x_{i-1}) + \Delta S_{3F}(x_i) - \Delta S_{3R}(x_i) \quad (11)$$

Eq. (11), which is same formula as Eq. (5), means that both of the compensation values h_a and ϕ are eliminated by the numerical calculations described above.

On the other hand, rotational error motion of the sensor head e_θ can be calculated using Eqs. (1) to (3) as expressed by Eqs. (12) and (13).

$$e_\theta(x_i) = \frac{S_{3F}(x_i) - S_{3C}(x_i) - g(x_{i+1}) + g(x_i) + (i-1)\alpha}{d} = \frac{\Delta S_{3F}(x_i) - \Delta g(x_i) + (i-1)\alpha}{d} \quad (12)$$

$$e_\theta(x_i) = \frac{S_{3C}(x_i) - S_{3R}(x_i) - g(x_i) + g(x_{i-1}) + (i-1)\alpha}{d} = \frac{\Delta S_{3R}(x_i) - \Delta g(x_{i-1}) + (i-1)\alpha}{d} \quad (13)$$

The inclination angles of the sensor head at starting point x_0 of shape measuring and at finishing point x_n are obtained from Eqs. (12) and (13), respectively as expressed by Eqs. (14) and (15). The difference $\Delta e_\theta(x_n)$ in inclination angles

between at x_0 and x_n is calculated by Eq. (16). On the other hand, the difference can be measured by some inclination sensor. By using the measured value instead of $\Delta e_\theta(x_n)$ in Eq.(16), α can be calculated and the shape profile is obtained with compensation of quadratic error component due to the zero position error between sensors.

$$e_\theta(x_0) = \frac{\Delta S_{3F}(x_0) - \Delta g(x_0) - \alpha}{d} \quad (14)$$

$$e_\theta(x_n) = \frac{\Delta S_{3R}(x_n) - \Delta g(x_{n-1}) + (n-1)\alpha}{d} \quad (15)$$

$$\begin{aligned} \Delta e_\theta(x_n) &= e_\theta(x_n) - e_\theta(x_0) \\ &= \frac{\Delta S_{3R}(x_n) - \Delta S_{3F}(x_0) - \Delta g(x_{n-1}) + \Delta g(x_0) + n\alpha}{d} \end{aligned} \quad (16)$$

MEASURING EXPERIMENT

Figure 4 shows the experimental measuring equipment to ascertain the principle described above. The X and Y axis were taken on the horizontal plane. Three electrical comparators were used for the displacement sensors. The working range and repeatability of the sensors were $\pm 25\mu\text{m}$ and $0.3\mu\text{m}$, respectively. The distance between sensors d was 50mm. The sensor head was mounted on an adjustment unit which is a stack of a small rotary table above a fine adjustment table both of which are driven by stepping motors. The positioning resolutions of the tables were $4.0 \times 10^{-5}\text{rad}$ and $0.01\mu\text{m}$, respectively. The adjustment unit was mounted on a scanning table driven by a ball screw and a stepping motor. The table has periodic error motion the amplitude of which is about $6\mu\text{m}$ corresponding to the pitch of ball screw. The distance between the test piece surface and the tip of the center sensor h_a and rotational angle φ of the sensor head were adjusted by driving the stepping motors as mentioned hereunder. When the center sensor moves in the direction of X-axis to a measuring position x_i from the previous

position x_{i-1} , h_a is adjusted so that the output signal of the sensor approximately coincides with the central value of working range of the sensor. Subsequently, φ is adjusted so that the difference of output signals of front and rear sensors is reduced to be less than $10\mu\text{m}$.

The inclination angles of sensor head $e_\theta(x_0)$ and $e_\theta(x_n)$ were measured by autocollimator as shown in Figure 4. When the inclination exceeds the working range of autocollimator, it was measured by the rotary table. The rotational angle given to the rotary table when the autocollimator reading becomes zero was considered to be the inclination angle of the sensor head. In this case, accuracy of angle measurement is lower than that measured by autocollimator.

A glass scale with smooth surface was used for the test pieces. The length, width and thickness of the scale were 950mm, 22mm and 4.5mm, respectively. The scale was supported at both ends with a distance of 916mm. To produce curved surfaces, a deflection Δl was given at the center between the supporting points of the scale by a micrometer.

Figure 5 shows the fluctuation of measured shape profiles of the middle part of test piece without any compensation when the deflection is approximately 1mm. The large periodic error motion of scanning table was eliminated. Figure 6 shows the magnified shape fluctuation from the average profile of them. The width of fluctuation was about $14\mu\text{m}$. The quadratic error components due to the change in zero position error between sensors seem to be superimposed on the profiles. Figure 7 shows the fluctuation after the compensation of quadratic error components by measuring the difference between the inclination angles of sensor head by autocollimator. The width of fluctuation decreased to about $0.30\mu\text{m}$ by the compensation. This fluctuation remaining is considered to be the intrinsic measuring error of the equipment.

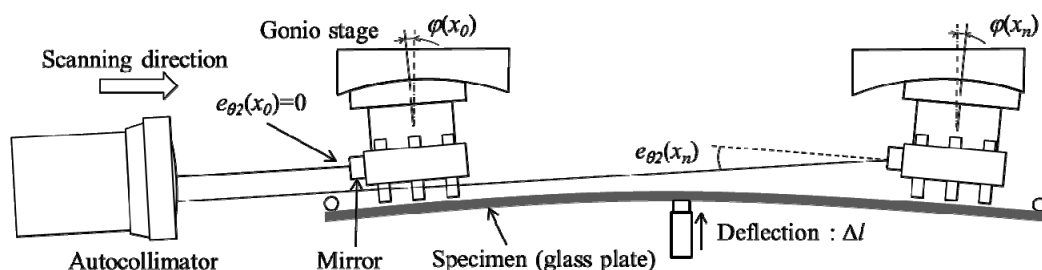


Figure.4 Experimental measuring equipment

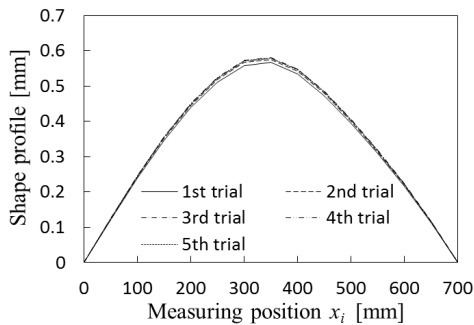


Figure 5 Shape profiles of the test piece ($\Delta l=1\text{mm}$)

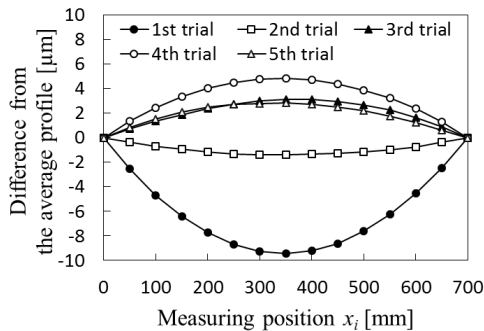


Figure 6 Fluctuation of shape profiles ($\Delta l=1\text{mm}$)

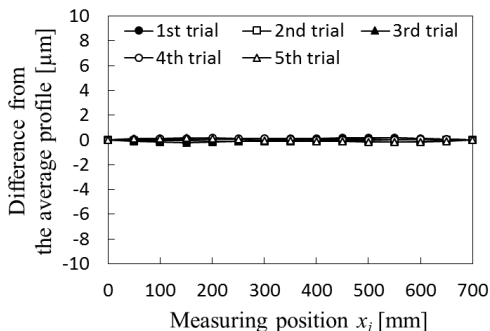


Figure 7 Fluctuation of shape profile with compensation of quadratic error component ($\Delta l=1\text{mm}$)

The shape fluctuation from the average profile without compensation when the deflection was approximately 3.5mm was shown in Figure 8. The width of fluctuation was about 6 μm . Figure 9 shows the fluctuation after the compensation of quadratic error components. The width of fluctuation decreased to about 1.2 μm by the compensation. In this case, as the curvature of test piece was larger than that in Figure 6, the curved surface did not enter in the working range of three sensors simultaneously. Therefore, zero positions of the sensors were consciously shifted so that all sensors can

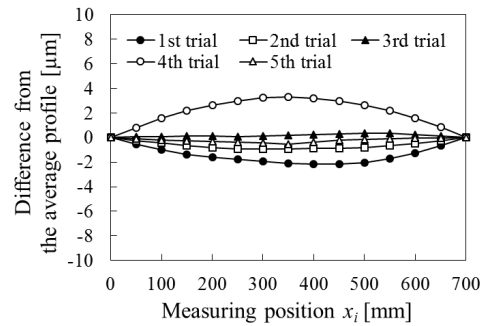


Figure 8 Fluctuation of shape profiles ($\Delta l=3.5\text{mm}$)

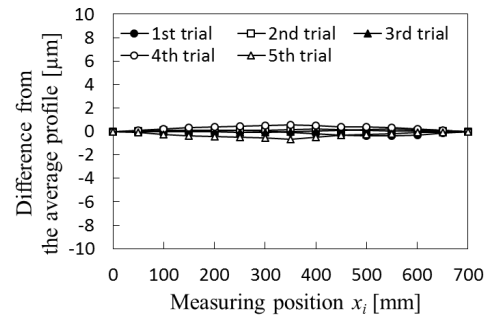


Figure 9 Fluctuation of shape profile with compensation of quadratic error component ($\Delta l=3.5\text{mm}$)

simultaneously detect the displacement to the test piece surface. The larger fluctuation width than that in Figure 7 is due to the lower measuring accuracy of inclination angle by rotary table.

SUMMARY

A design concept of new surface shape measurement of large curved surface is proposed using improved three-point method. The measuring experiment based on the concept suggests that the method has enough measuring accuracy of 0.3 $\mu\text{m}/700\text{mm}$ even by using scanning table with considerably large error motion.

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MINIMIZING TASK-SPECIFIC UNCERTAINTY IN CMM-BASED FREEFORM OPTICS METROLOGY

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INTRODUCTION

All measurements of surface form involve a comparison between the form of the part under test, and a reference. In coordinate measuring machine (CMM) measurements, this reference is the machine geometry. The measurement results typically show a height map representation of the part, but these also contain error contributions from the machine. This paper demonstrates simulations of an in-situ self-calibration technique, Shift-Rotation, which minimizes task-specific uncertainty in freeform measurements, by separating machine errors from part surface measurements. This calibration technique has been in use in interferometry for testing optical surfaces [1, 2, 3, 4], and the idea is here, extended to measurements from CMMs.

Some potential uncertainty sources are associated with this error separation technique, and these uncertainties can be expressed as height maps that show the spatial distribution of the uncertainty within the aperture. Task-specific uncertainty analyses, using Monte Carlo, will be used to evaluate these uncertainty contributions.

SIMULATIONS OF THE SHIFT-ROTATION TECHNIQUE

In Shift-Rotation, form measurements are considered as comprising rotationally variant (RV) and rotationally invariant (RI) components of the part and machine errors. These will be represented as P_{RV} , P_{RI} , M_{RV} , and M_{RI} respectively. The technique will be used to produce four output height maps: Output P_{RV} , Output P_{RI} , Output M_{RV} , and Output M_{RI} . In this paper, a combination of the Output P_{RV} and Output P_{RI} , will be used to represent a calibrated measurement of the part, free from the machine's error contributions.

In these simulations, combinations of Zernike polynomials are used to represent the part and machine error components. These simulations are deliberately performed with large form deviations and the amplitudes of the simulated machine errors are also deliberately large.

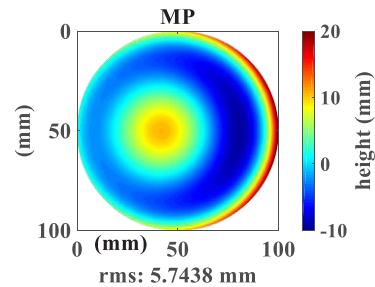


FIGURE 1. A simulated measurement of a freeform surface.

Figure 1 represents a measurement of a freeform part with a 100.1 mm diameter circular aperture. This measurement result depicts the freeform surface, but also includes errors from the measuring machine. In order to separate the components in the above height map, the “rotation” portion of the Shift-Rotation technique will be used to solve for the RV components of the part and machine, while the “shift” portion solves for the RI components.

The next two sections show how measurement results, obtained after multiple part rotations and translations, can be used to obtain RV and RI components of the part and machine.

Separation of the Rotationally Variant Components

To solve for the RV components of the part, measurements of the part are made after the part is rotated (about the optical axis) to N positions, separated by $360^\circ/N$. The average of these measurements would then contain none of

the RV components of the part, except those that are harmonics of $360^\circ/N$ [5]. The number of rotations required would be higher than the highest angular order expected to be of significant amplitude on the part. Since these simulated height maps were generated with a highest Zernike angular order of 3Θ , four rotations are sufficient to solve for the RV components. Simulated measurement results, from four rotated positions of the part, are shown in Figure 2.

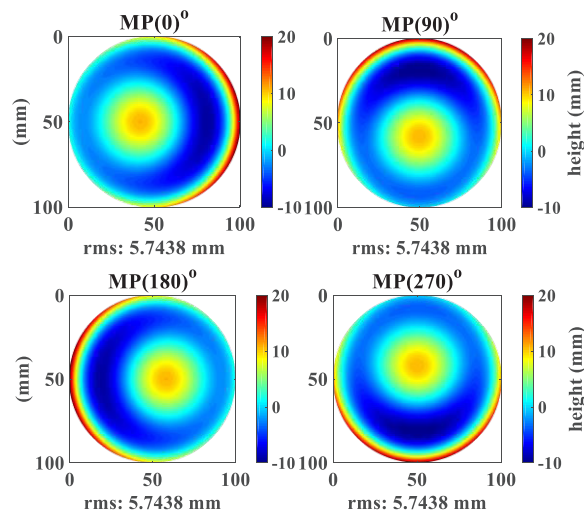


FIGURE 2. Measurements after four rotations of the freeform surface.

The average of the height maps in Figure 2, is a height map (let us refer to as Mean_P) without the RV components of the part. The RV component of the part, removed by the averaging process, can be obtained after a subtraction and is shown in Figure 3.

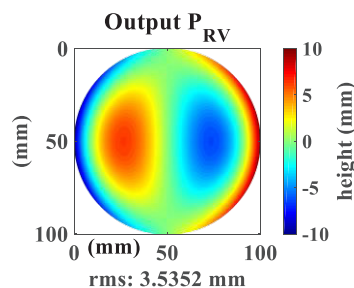


FIGURE 3. The RV component of the freeform part.

Output P_{RV} represents one of the four desired separated components and should be stored for later use. No additional measurement is required

to solve for the RV component of the machine errors. This can be obtained by repeating the same rotation and averaging process on Mean_P, and is shown in Figure 4.

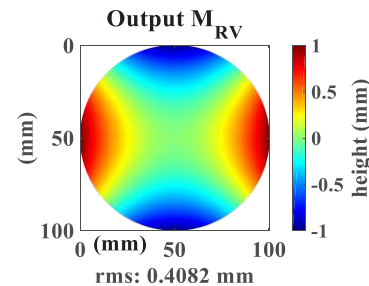


FIGURE 4. The RV component of the machine errors.

Output M_{RV} represents the second of the four desired separated components, and should also be stored for later use. Now that the RV components have been separated, the next section can be used to separate the RI components.

Separation of the Rotationally Invariant Components

Separation of the RI components requires measurements of the part, before and after the part has been laterally shifted by a known translation distance, with respect to the machine. For example, Figure 5 (left) shows a simulated measurement of the part in an initial position, and after the part was translated 2 mm to the left, its measurement was that depicted in Figure 5 (right).

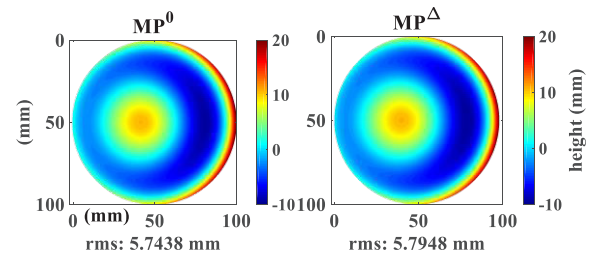


FIGURE 5. Measurement of the part in an initial position (left). Measurement after the part was shifted 2 mm to the left (right).

Since the RV components of the part and machine have already been obtained from the previous section, these can be subtracted (in their respective orientations) from MP^0 and $MP^Δ$. Then, a difference between the resulting maps

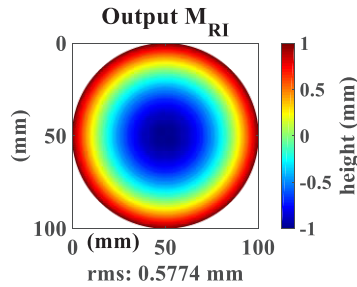


FIGURE 9. The RI component of the machine errors.

In solving for the above RI components, only one lateral translation was used. If only one translation is to be used, the translation distance should be small compared to the aperture size, limited by signal to noise considerations. If a large translation distance is used, or if the solved radial profile (Figure 7) is not a continuous curve, two or three lateral translations might be necessary to solve for the RI components. In this case, the linear system described in Table 1 would be a combination obtained from the multiple lateral translations.

The Calibrated Measurement of the Freeform Part

To summarize, if the height map in Figure 1 represents an uncalibrated measurement; the sum of the Output P_{RV} and Output P_{RI} height maps can represent a calibrated freeform measurement with a minimized uncertainty because the RV and RI machine errors have been removed. This calibrated measurement of the freeform part is shown in Figure 10 below.

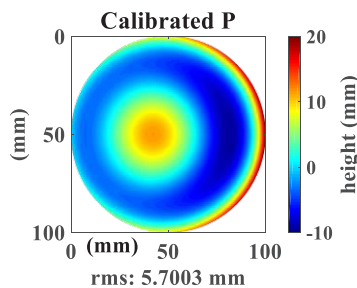


FIGURE 10. A calibrated measurement of the freeform surface, without the RV and RI machine errors.

The next section evaluates task-specific uncertainties associated with the calibrated measurement of the freeform part.

MONTE CARLO UNCERTAINTY ESTIMATIONS

The Monte Carlo method may be used, in uncertainty analyses, to evaluate uncertainty contributions from various sources [7]. Some potential uncertainty sources associated with the error separation process are listed in Table 2.

TABLE 2. Potential sources of uncertainty associated with the Shift-Rotation.

Uncertainty sources	Measurand	Output uncertainty
Rotation angles	Evaluated by using the calibrated freeform height map	Task-specific uncertainty height map
Translation distance		
Moving axis of rotation		
CMM parametric errors		
Electronic noise and vibrations		

The Monte Carlo simulation is performed by building a model that generates many possible outcomes, which could arise from the variation of an input parameter. Here, the Monte Carlo simulation will be illustrated by an example where the rotation angle is used as the input parameter.

In the rotation technique shown in the previous sections, it was assumed that each rotation angle was an exact integer multiple of 90° . This led to a height map representation of the calibrated freeform (Figure 10), and this height map was further represented by an amplitude parametric surface specification ($\text{rms} = 5.7003 \text{ mm}$). In using the Monte Carlo, instead of assuming we have perfect rotations, we presume that each rotation angle can fall anywhere within some distribution. In the flowchart shown in Figure 11, uniform distributions with upper and lower limits of $\pm 0.05^\circ$ are used around each nominal rotation angle.

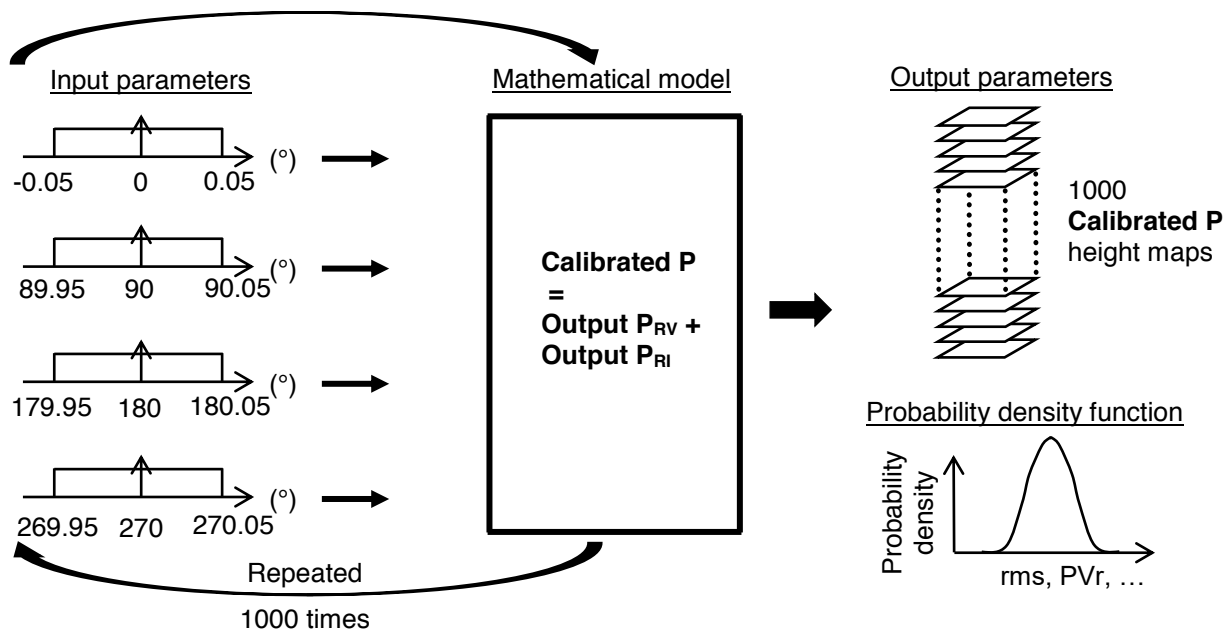


FIGURE 11. A flowchart of the Monte Carlo process, with rotation angle as the input parameter.

After the process is repeated a large number of times, such that each trial would have random angles from the distributions, an array of possible outcomes of the freeform part measurement would be obtained. The average of the height maps of the many possible outcomes stored in the array, gives an estimate of how well the Monte Carlo has converged, while the standard deviation across the array, leads to a combined standard uncertainty.

After this process was applied to the simulated height maps in the previous section, the Monte Carlo converged to the height map shown in Figure 12 (left). The uncertainty in the measurement of the measurand, as a result of an uncertainty in the rotation angle, is shown as the height map in Figure 12 (right). This uncertainty is a height map, the same size and data density as the measurand, and shows the spatial distribution of the uncertainty associated with the measurement of the measurand. The amplitude parametric representation of the possible measurement outcomes in the array, could then be expressed as the probability distribution in Figure 13, reported with a coverage factor of 2.

The uncertainty map shown in Figure 12 (right), was obtained as a result of uncertainty in the rotation angles, only. A combined standard uncertainty can be obtained by applying the

same Monte Carlo process to other uncertainty sources, for example, those listed in Table 2.

Actual CMM measurements will contain spatial and computational errors [8] and a number of methods are available for evaluating these machine errors [8, 9, 10, and 11]. With the Shift-Rotation technique, it is unnecessary to measure these machine errors which include the parametric errors (21 D. o. F), since the separated RV and RI components of the machine errors can be added to give a height map representation of the task-specific machine error contributions. These machine error contributions can be separated out, on a task-specific basis, anywhere in the machine's measurement volume. Also, since this calibration method requires stability only over the measurement cycle time, effects of machine drifts are reduced to the timescale of the measurement rather than the interval between machine calibrations.

The next step involves experimental validation of the above calibration technique. Rotation and translation measurements, similar to those described in this paper, will be made on a Zeiss F25 CMM and/or a Mahr MarSurf LD 260 surface profiler. From these measurements, a calibrated freeform part measurement would be obtained, as well as its task-specific uncertainty height map.

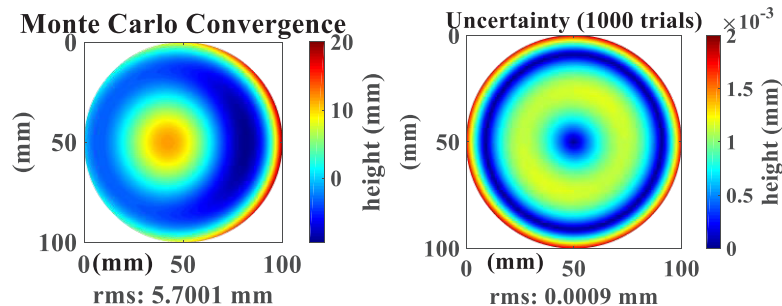


FIGURE 12. The convergence of the Monte Carlo to the Calibrated P height map (left). The uncertainty in the best estimate measurement (right).

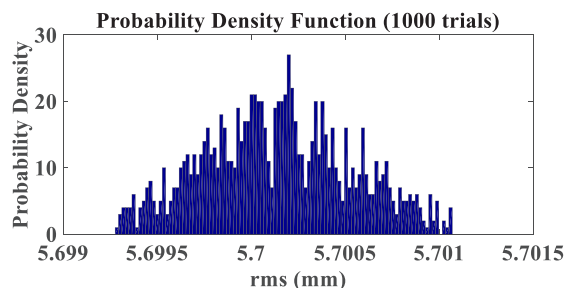


FIGURE 13. Amplitude parametric representations of the 1,000 trials in the Monte Carlo, reported as: $\text{rms} = 5.7001 \pm 0.0008 \text{ mm}$ ($k = 2$).

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GAGE REPEATABILITY AND REPRODUCIBILITY STUDY OF A HEMISPHERICAL SHELL

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INTRODUCTION

Currently LANL's dimensional inspection on hemispherical shells (hemi-shells) is conducted on uniquely designed rotary contour machines. The rotary contour machines are designed to measure the radial wall thickness of a hemi-shell. These machines have various points of failure, no manufacturer support and age leading to production down time. LANL has begun to explore and invest in more modern technologies to support future endeavors. Coordinate Measuring Machines (CMMs) are a staple in manufacturing and production engineering and utilize newer technologies, require less operator involvement and measure a variety of different shaped parts, with very good precision and accuracy.

This research uses a Gage Repeatability and Reproducibility (GR&R) study, with a Precision Measuring Machine (PMM-C), to investigate the measurement process variation during hemi-shell radial wall thickness measurements and is analyzed for total process variation, as well as the Precision-to-Tolerance (P/T) Ratio [1]. The GR&R will also indicate a 4:1 measurement uncertainty ratio from government specification 9900000 [2], which states that the measuring instrument used must measure as low as 25% of the customer performance specifications.

GAGE R&R Background

A GR&R study is a statistical tool that measures the amount of variation in the measuring system from the measuring device (gage) and the people (operator) taking the measurements [3]. It indicates the consistency and stability of the measuring system and operator and can also determine and quantify where the process variability exists.

The ANOVA statistical method will be the method of choice for the GR&R study for accuracy purposes [3,4,5,6,7]. The repeatability and

reproducibility, along with the (P/T) ratio, will indicate variability associated with the measurement process and if the PMM-C is capable of meeting customer performance specifications.

Analysis of Variance

ANOVA is a statistical method using the approach of Analysis of Variance. ANOVA is a collection of statistical models, and their associated procedures, in which the observed variance is partitioned into components of different sources of variation. ANOVA GR&R considers several sources that affect the measuring system: operators, testing methods, part setup, performance specifications and the measuring system itself. ANOVA GR&R methodology is more accurate because it breaks down the reproducibility portion into operator interaction and operator-by-part interaction [7]. This can be seen in FIGURE 1.

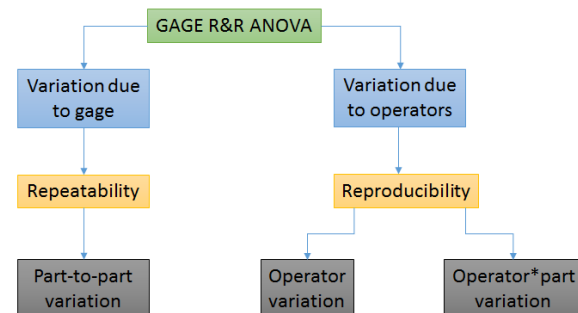


FIGURE 1. ANOVA GR&R Method

A Gage R&R is a two-factorial experiment, where r is repeatability and R is reproducibility. Each factor can have a sub-level(s) and have direct correlation to the factors being assessed. Total measurement process variation is quantified with total GR&R variation, and variations from operator, operator-part interaction and part-to-part. The Gage R&R linear model is described below:

$$Y_{ijk} = \mu + \tau_i + \beta_j + (\tau\beta)_{ij} + \epsilon_{ijk} \begin{cases} i = 1, 2, \dots, a \\ j = 1, 2, \dots, b \\ k = 1, 2, \dots, c \end{cases} \quad (1)$$

Where Y_{ijk} is equal to the i^{th} part from the j^{th} operator of the k^{th} measurement; μ is the mean of all measurements taken by all operators; τ_i is the effect of the i^{th} level of factor $\mathbf{r}$; β_j is the effect of the j^{th} level of factor $\mathbf{R}$; $(\tau\beta)_{ij}$ is the effect of the interaction between $\mathbf{r}$ and $\mathbf{R}$, and ϵ_{ijk} is the random error component having a $N(0, \sigma^2)$ distribution.

This linear model can then be used in a sum-of-squares (SS) to quantify measurement process variability and has four sources of variation for total variation: sum-of-squares for $\mathbf{r}$ (SS_r), sum-of-squares for $\mathbf{R}$ (SS_R), sum-of-squares for interaction (SS_{rR}) and sum-of-squares for error (SS_E), which is summed up for total sum-of-squares (SS_T). SS_T can be seen in Eq. (2) and a symbolic version in Eq. (3).

$$\begin{aligned} & \sum_{i=1}^a \sum_{j=1}^b \sum_{k=1}^n (y_{ijk} - \bar{y}_{...})^2 \\ &= \sum_{i=1}^a \sum_{j=1}^b \sum_{k=1}^n [(\bar{y}_{i..} - \bar{y}_{...}) + (\bar{y}_{.j.} - \bar{y}_{...}) \\ &+ (\bar{y}_{ij.} - \bar{y}_{i..} - \bar{y}_{.j.} + \bar{y}_{...}) + (y_{ijk} - \bar{y}_{ij.})]^2 \\ &= bn \sum_{i=1}^a (\bar{y}_{i..} - \bar{y}_{...})^2 \\ &+ an \sum_{j=1}^b (\bar{y}_{.j.} - \bar{y}_{...})^2 \\ &+ n \sum_{i=1}^a \sum_{j=1}^b (\bar{y}_{ij.} - \bar{y}_{i..} - \bar{y}_{.j.} + \bar{y}_{...})^2 \\ &+ \sum_{i=1}^a \sum_{j=1}^b \sum_{k=1}^n (y_{ijk} - \bar{y}_{ij.})^2 \end{aligned} \quad (2)$$

$$SS_T = SS_r + SS_R + SS_{rR} + SS_E \quad (3)$$

The repeatability and reproducibility for the GR&R are calculated from the standard deviations of the part-to-part, operator and operator-part interaction variations. The formulas can be seen in Eq. (4) and Eq. (5).

$$S_{ijk} = \sqrt{\frac{1}{k-1} \sum_{k=1}^k (Y_{ijk} - \bar{Y}_{ij.})^2} \quad (4)$$

$$\bar{Y}_{ij.} = \frac{1}{k} \sum_{k=1}^k Y_{ijk} \quad (5)$$

Where S_{ijk} is the standard deviation of the i^{th} part from the j^{th} operator of the k^{th} measurement. $(k-1)$ is the degrees of freedom. Squaring the standard deviation will give the variation of the particular measurement.

The (P/T) ratio addresses the percent tolerance variation and determines the suitability of the measurement system. The percent tolerance must be $\leq 10\%$ of measurement process variability for critical response variables, between 10% and 30% variability for non-critical response variable and unacceptable for $> 30\%$ of process variability. LANL's measurement processes are defined as $\leq 10\%$ for critical response variables. The (P/T) ratio is modeled as below:

$$(P/T) = \frac{K * \sigma_{R\&R}}{USL - LSL} \quad (6)$$

The K value, $K = 5.15$, corresponds to the limiting value of the number of standard deviations between bounds of a 95% tolerance interval that contains at least of 99% a normal population [6]. The $\sigma_{R\&R}$ is the total repeatability and reproducibility variation in the measurement process. USL and LSL are the upper and lower specification limits of the part under measurement.

EXPERIMENTAL TESTING

Measurement Program

Data is collected every two degrees from 0° to 88° , including 0° , which results in 45 data points in the polar direction and measured every 30° degrees in the azimuth direction which will equate to 12 polar bands of data. The same sampling strategy is done for both the inner contour and outer contour of the hemi-shell. 540 data points are measured from each contour and a total of 1080 points is collected. However, since the radial wall thickness is determined from both the inner and outer contour, the radial wall thickness measurement is a calculation and will produce 540 total measurement points. FIGURE 2 shows the measurement directions for both inner and outer contours of the hemi-shell.

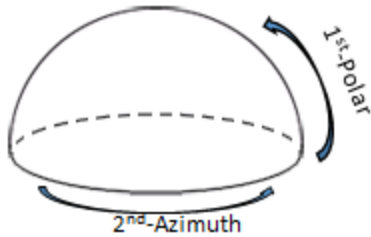


FIGURE 2. Measuring Direction of Hemi-shell

The program is conducted via a scanning routine and not a touch trigger routine because of shorter time duration needed during the measuring process. Though touch trigger probing is more accurate, time becomes a factor for the amount of data recorded from each hemi-shell. If the same scanning routine were implemented with touch trigger probing, the measuring routine would be significantly longer, thus violating one of the purposes for researching this project, time constraints.

Experimental Setup

The hemi-shells are mounted on a custom fixture to collect all measurements. The mounting fixture is fabricated from a solid piece of aluminum. To hold the hemi-shells on the fixture, unique adapters, known as rounding rings, are fitted and mounted on the fixture. Once all fixtures and mounts are fastened, the mounting fixture is clamped to the granite table by way of bolts and toe clamps. FIGURE 3 shows a mock measurement setup, with a mock hemi-shell.



FIGURE 3. Measurement Setup (Unclassified)

Before measuring the hemi-shell, the machine coordinate system and part coordinate system must be set, to reference all data points. After coordinate setup the measuring process can begin. FIGURE 4 and FIGURE 5 show the measurements of the inner and outer contours of

a hemi-shell, used to calculate the radial wall thickness.

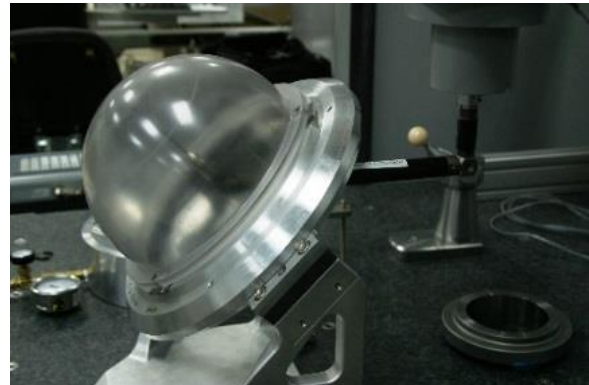


FIGURE 4. Inner Contour (Unclassified)

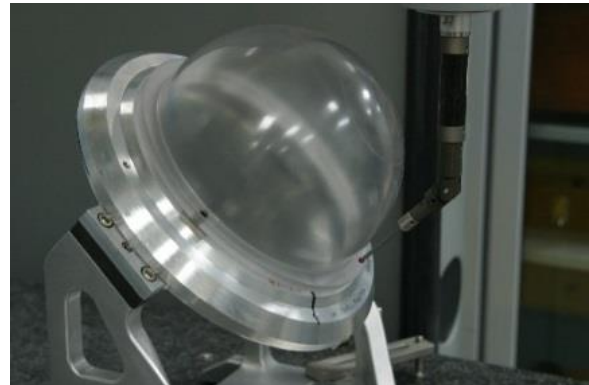


FIGURE 5. Outer Contour (Unclassified)

RESULTS

The Gage R&R used six hemi-shells, two operators and three trials for each hemi-shell and produces a total of 36 measurements. Only the measurements from the pole, midpoint and equator of the hemi-shell will be analyzed, as these are some of the more critical locations of the machining process. To determine how much variation contributions there was in each measured location, the percent variation of each source of the GR&R is stated. The total, for each measured location, totals to 100%. TABLE 1 shows the percent component variation, of the measuring process, for each measured location.

TABLE 1. Variance Components for Pole, Midpoint and Equator

Variance Components (%)	Repeat.	Reprod.	P-to-P	OpPart Int.	Part
Pole	67.03	0.00	32.97	32.97	0.00
Midpoint	95.23	0.98	3.79	3.79	0.00
Equator	96.85	1.68	1.46	1.46	0.00

For better visualization, the percent contribution for each measured location is plotted. FIGURE 6,

FIGURE 7 and FIGURE 8 show the percent contribution for each source of each measured location.

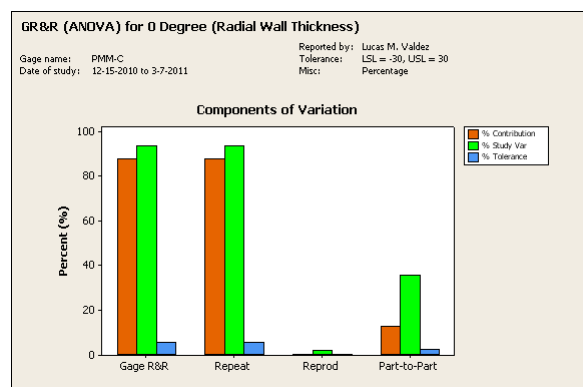


FIGURE 6. Component Variation at Equator (0°)

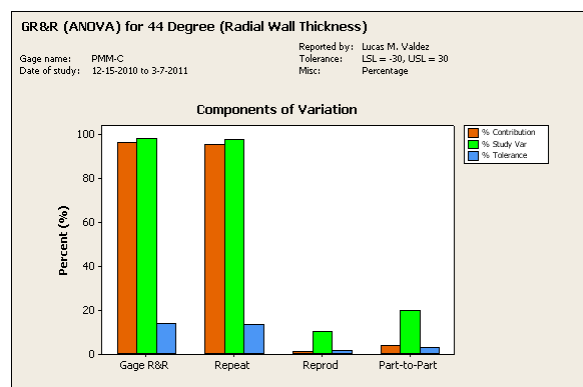


FIGURE 7. Component Variation at Midpoint (45°)

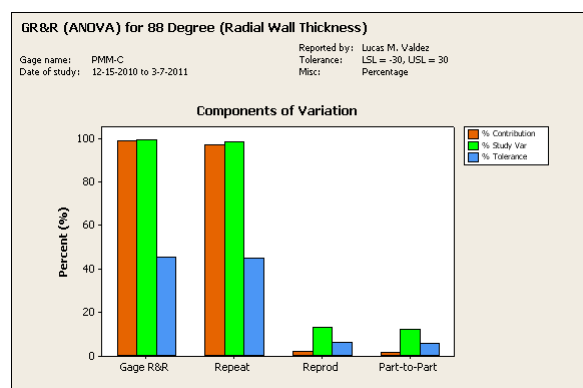


FIGURE 8. Component Variation at Equator (88°)

The majority of the process variation is from the repeatability, which indicates most of the variation comes from the PMM-C. This observation can be seen for all three measured locations. It should be noted that the PMM-C used in this study was close to its year-end calibration cycle. This could be a potential influence on the measured results.

Another issue is that this machine was not interim checked very frequently, which could have identified potential errors in the machine geometry before this initial investigation into the measurement process capabilities.

To see if the variability of the sources in the process have a significant effect on the measured locations, the (P/T) ratios are determined. These values will tell us if the PMM-C can measure the hemi-shells to within the needed accuracy. To reiterate, the (P/T) ratio is calculated for the total variation, GR&R and the 990000 4:1 government specification. TABLE 2 shows the precision-to-tolerance ratio of the measurements.

TABLE 2. Precision-to-Tolerance Ratios

P/T Ratio	Pole (0°)	Midpoint (45°)	Equator (88°)
% Total Variation	6.10	1.37	4.54
% GR&R	4.99	1.35	4.51
% 4:1 Ratio	4.99	1.34	4.47

The P/T ratios meet the $\leq 10\%$ of process variability for critical response variables. To summarize, the results show that at each measurement location, the PMM-C and inspector variability are acceptable for measuring a hemi-shell. The 4:1 measurement uncertainty ratio is well under the 4: 1 value for measuring equipment used in LANL's production process. Overall, the measuring process of a hemi-shell on a PMM-C is acceptable for WR quality work done at LANL.

CONCLUSION AND FUTURE WORK

The Gage R&R produced valid results for all three measurement locations, both in percent variation and P/T ratios. These results provide validation that the PMM-C can measure hemi-shells with limited variation. Future work is to implement more GR&R's on more measuring equipment in LANL inventory. Also, calibrating the PMM-C and repeating the GR&R to compare the results would provide better feedback to the standards and calibration laboratory. Further analysis into interim checking the CMM/PMM's in the inventory are underway to ensure no misrepresented data is published to the customers. Optimization of the entire process is currently underway and results will be published in the near future.

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A STUDY OF UNCERTAINTIES FROM WORKPIECE TILT IN DIMENSIONAL COMPUTED TOMOGRAPHY

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Previous work [1] aimed at finding optimal specimen orientations within cone-beam X-ray computed tomography (CT) systems by comparing dimensional measurements using CT and a coordinate measurement machine (CMM). An optimal placement was assumed to minimize deviations between the two measurements. Alternatively, in the work presented in [2, 3, 4], a “Dreamcaster” simulation tool was introduced to estimate the optimal orientation of the specimen before the CT scan based on a 3D CAD model. The simulation was performed using ray casting and Radon-space analysis to minimize both beam-hardening and Radon-space shadow-zone artifacts. In this work, a comparison between the experimental results for measurement deviations and simulated data from the Dreamcaster is presented.

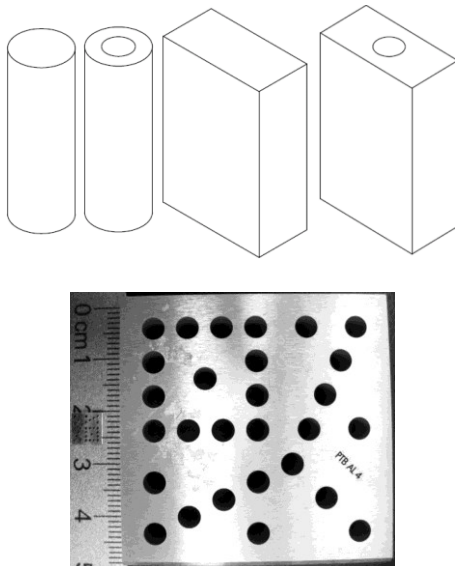


FIGURE 1. Examined specimens [1, 5].

A study was conducted at the University of North Carolina Charlotte (UNCC) to investigate uncertainties from workpiece tilt in dimensional measurements performed with CT [6]. Four specimens with simple geometrical shapes were

examined: a rectangular solid block, a rectangular hollow block, a solid cylinder, and a hollow cylinder (see Figure 1). The specimens' nominal dimensions are: length of 60 mm, blocks' base dimensions of 20 mm × 40 mm, 20 mm radius for the solid cylinder, and a 10 mm diameter of drilled holes. The workpieces were manufactured from aluminum and poly-methyl methacrylate (PMMA) thermoplastic. In addition, two hole plates of different sizes were tested (shown in the bottom of Figure 1). The larger hole plate is made of aluminum and the smaller hole-plate is made of steel. Each hole-plate has 28 drilled through-holes. The hole-plates were provided by National Metrology Institute of Germany (PTB, Physikalisch-Technische Bundesanstalt). Due to limitations of space, the analysis of CT results presented in this paper is restricted to the solid block and the steel hole-plate. A more comprehensive analysis that includes CT measurement results for the other geometrical shapes introduced in Figure 1 will be presented in reference [6].

EXPERIMENTAL SETUP

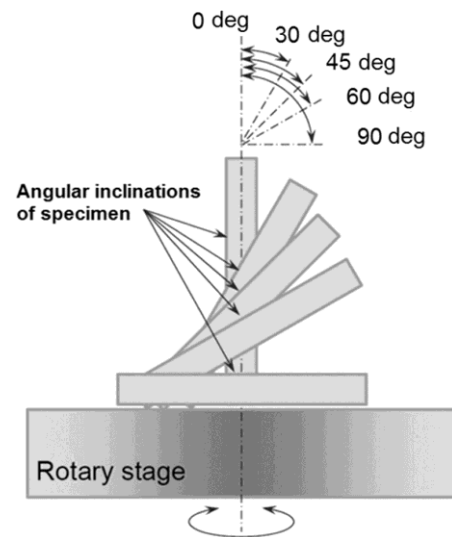


FIGURE 2. Experimental setup of reference [1].

To enable CT scans of the workpieces at different angular inclinations, the setup shown in Figure 2 was used. The study is limited to CT configurations of cone-beam type, which are the most widely used configurations for dimensional metrology. The measurements were performed with a Zeiss Metrotom 800 CT system housed within the Center for Precision Metrology at UNCC between 2012-2014. Additional data was collected at the Metrology Services Center from Carl Zeiss Industrial Metrology, LLC at Brighton, MI (US). The extraction of dimensional data was performed with the Calypso 5.6 software [7], and compared to CMM reference values obtained with a Zeiss Micura CMM system.

SIMULATION PROCESS

Simulation was carried out based on 3D CAD models of the specimens using the Dreamcaster tool [2, 3, 4]. Traversal-path lengths of X-rays through a specimen were computed by simulating the cone-beam CT scanning process using ray casting. Subsequently the percentage of surface area that is lost or underrepresented in the Radon-space was estimated by testing triangles of 3D CAD model against the Tuy-Smith's condition of sufficiency [8, 9]. As the result a total of three parameters were computed for every angular inclination of each tested specimen: the average traversal-path length ($avg(TPL)$), the maximum traversal-path length ($max(TPL)$), and the percentage of surface area lost or underrepresented in Radon space ($pct(SURS)$). These parameters are functions of the specimen's geometry as well as its orientation and location in the cone-beam CT scanner. For the purpose of this study, the axial tilt angle θ , measured between the workpiece's main axis and the CT rotating axis (Z) (see the inset of Figure 3), is the only variable. So, the Dreamcaster simulation is performed as a function of the angle θ . Other geometrical variables were set with values that are identical to the ones used in the experimental CT setup. For comparability, each parameter $p(\theta)$ computed from the Dreamcaster was normalized based on its full range of values for a set of angles θ , with normalization given by

$$P = \frac{p - \text{Min}(p)}{\text{Max}(p) - \text{Min}(p)} \quad (1)$$

Normalized values of all three simulated parameters are combined into a single global parameter, D_{sim} , using an equally weighted sum:

$$D_{sim}(\theta) = \text{Avg}(TPL) + \text{Max}(TPL) + \text{Pct}(SURS) \quad (2)$$

The first two terms, $\text{Avg}(TPL)$ and $\text{Max}(TPL)$, are related to beam hardening effects as high traversal-path lengths of X-ray radiation might cause distortions or artifacts in the CT data [2, 10, 11, 12], and the last term, $\text{Pct}(SURS)$, refers to the percentage of surface area underrepresented in the Radon-space due to incompleteness of the cone-beam acquisition geometry [2, 8, 9, 13].

PRELIMINARY RESULTS

The presented measurement values are the averaged results of three separate measurement repetitions that were performed at the standard reference temperature $T = 20^\circ\text{C}$.

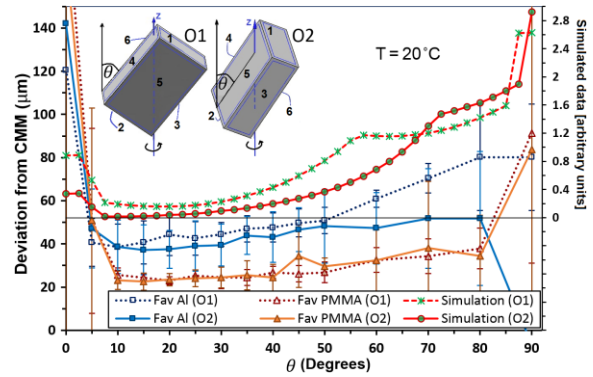


FIGURE 3. Mean deviation of CT measurements of flatness (F) averaged over all planar faces. Vertical bars represent standard deviations.

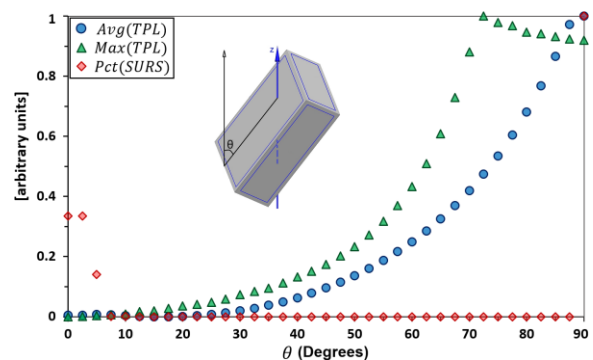


FIGURE 4. Data obtained from the Dreamcaster simulation for the solid blocks in orientation O2.

Solid blocks

In Figure 3, the parameter $D_{sim}(\theta)$ for the solid blocks obtained from the Dreamcaster simulation is compared graphically against the experimental

results of mean deviations for the measurements of flatnesses (F) averaged over the planar blocks' faces. Two different orientation configurations, O1 and O2 (shown in Figure 3), were tested for θ tilt variations between 0 to 90 degrees. To find the optimal orientations for CT scanning, the experimental measurements were performed on unfiltered results. Neither smoothing filters nor outlier elimination for extracted dimensional data have been used to avoid filtering out data that would be potentially useful in the discrimination phase of optimization. From the results presented in Figure 3 it can be seen that the simulation data show a trend comparable to the behavior of experimental CT measurement deviations: from a local maximum at $\theta = 0$ degrees, all numerical values start decreasing, reaching minimum values in the zone between 10 and 20 degrees, after which they start to increase slowly. After 55 or 60 degrees, the numerical values become quite large again reaching a maximum value at $\theta = 90$ degrees. Although values associated with simulated and experimental data respectively should demonstrate correlation when data from simulations and experiments agree, they are not directly or numerically comparable. The experimental results represent dimensional measurement deviations between CT and CMM in μm , but the simulated data represent values of $D_{sim}(\theta)$ generated by Eq. (2) and quantified in arbitrary units. Figure 4 shows details from the simulated data for the solid block in the orientation configuration O2 (referred to inset of Figure 3). From the numerical values associated to each term, it is found that $Pct(SURS)$ is mainly affected when θ is between 0 to 7.5 degrees and at 90 degrees, which in theory should correspond to unmeasured planes in the spatial domain that cannot be mapped into the Radon domain [2, 8, 9]. Now, $Avg(TPL)$ and $Max(TPL)$ are mostly constant from 0 to 45 degrees, but they start becoming larger after 50 degrees as the traversal-path lengths for X-ray radiation increase. Beam hardening effects associated with these long traversal-path lengths are prone to produce distortions in the CT datasets [2, 10, 11, 12]. For the orientation configuration O1 (refer to inset of Figure 3), the overall behavior for CT measurements on the solid blocks, exhibited by both experimental and simulated data, is basically the same as described above for the configuration O2, but with slightly larger deviations particularly in the metallic workpiece, as shown in Figure 3. Interestingly enough, from the simulated data as well as from the experimental results, at any particular tilt angle θ

between 10 and 65 degrees, the orientation O2 minimizes the influence on the CT measurements compared to the configuration O1. This is important for optimization of measurement and for minimizing uncertainty values of the CT results. As a result, O2 with the tilt angle between 10 and 20 degrees is the preferred configuration to be used for CT scan as it corresponds to the minimum deviations.

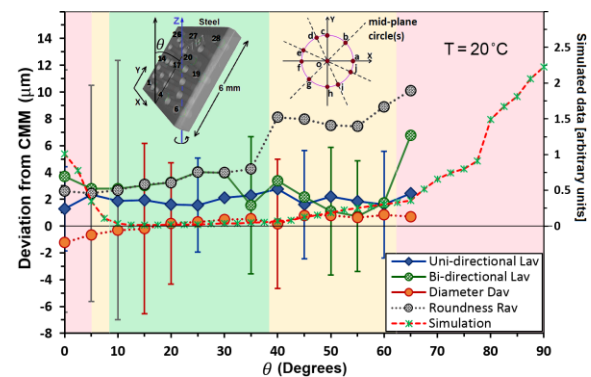


FIGURE 5. Mean deviations for dimensional CT measurements on the PTB steel hole plate. Vertical bars represent the expanded uncertainty of the CT measurements closest to CMM data.

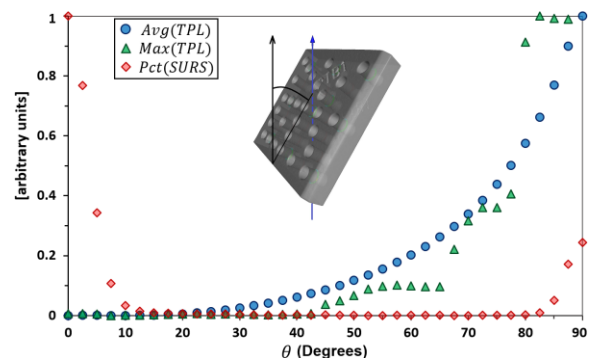


FIGURE 6. Data obtained from the Dreamcaster simulation for the PTB steel hole-plate artifact.

Steel hole-plate

Figure 5 shows the results of mean deviations for the CT measurements on the steel hole-plate. These are based on measurements of the ten circles labeled as 1, 4, 6, 14, 17, 19, 20, 26, 27, and 28 in Figure 5. The dimensional measurements are taken at the mid-plane between the top and bottom surfaces of the plate, using the circles of intersection between the mid-plane of the hole-plate and the holes as references. The measurement strategy follows reference [14, 15], which defines uni-directional distances (center-to-center) as measured

between the centers of two circles, and bi-directional distances (point-to-point) as measured between two points with the widest distance between two circles. To compute the deviation values shown in Figure 5, the dimensional measurements from CT used filtering (plus outlier elimination), a low-pass spline filter of 15 undulations per revolution. Also beam hardening correction (BHC) algorithms from the Zeiss Metrotom OS 2.4 software were applied to the CT raw data to correct for the strong X-ray beam hardening effects commonly present in metallic parts (particularly the high iron content, steel alloys). The data presented in Figure 5 shows that the optimal range of angular orientations (θ) for the PTB steel hole-plate is between 10 and 35 degrees (green zone). In this range discrepancy between CT and CMMs measurements is minimal for the majority of measured features (size dimensions, form, uni-directional and bi-directional lengths). The smaller deviations are obtained for measurements of the circles' diameters (computed from averaging around 850 data points on each circle) and also for uni-directional distances. The bi-directional lengths, which are based on two-point measurements, have larger deviations similar to measurements of form. For example, at $\theta = 20$ degrees, the mean deviation of CT measurements for roundness is $3.2\text{ }\mu\text{m}$; for diameter it is $0.2\text{ }\mu\text{m}$; for uni-length it is $1.6\text{ }\mu\text{m}$; and for bi-directional length it is $3.2\text{ }\mu\text{m}$. For some measured features deviation between CT and CMM measurements reaches their minima between 40 and 60 degrees (yellow zone), particularly for the bi-directional lengths, but for others, e.g., roundness, the mean deviation is relatively large ($7.4\text{ }\mu\text{m}$). In general, bi-directional length measurements have larger variations of deviations from CMM than the uni-directional ones, as seen in Figure 5, but it is for measurements of form, which are based on the two most extreme deviations from average, that the deviations between CT and CMM become larger, especially after a 35-degree tilt angle. The simulation, for example, predicts the larger deviations to be between 0 and 7.5 degrees and between 60 and 90 degrees (the 'red zones' in Figure 5). This is consistent with experimental results for the angles between 60 and 90 degrees, and partially with the measurement results between 0 and 10 degrees given the presence of larger uncertainties in that region. After 65 degrees, from the experimental trend shown, the deviations between CT and CMM are expected to grow, particularly for bi-directional

lengths and measurements of form. However, although there were CT scans collected between $\theta = 65$ and $\theta = 90$ degrees, performing dimensional measurements in these cases produced inconsistent results for the measurement software used (Calypso 5.6) due to the large number of CT distortion/artifacts present in the data after surface reconstruction. In that θ range the Dreamcaster tool predicts strong distortion/artifacts as can be seen from the simulated data in Figure 5, mainly related to beam hardening effects given the accentuated grow of $Avg(TPL)$ and $Max(TPL)$ (see Figure 6). Note that surfaces underrepresented in the Radon-space $Pct(SURS)$ is only significant in the regions from 0 to 7.5 degrees and between 85 to 90 degrees. The green zone that minimizes the deviations between the measurements with CT and CMM for the majority of dimensional measurements is between 10 and 35 degrees. It coincides with the optimal region of angular orientations predicted by $D_{sim}(\theta)$, as can be seen in Figure 5, and thus the trend between the Dreamcaster simulations and experimental data share similarities. The match between simulated and experimental data becomes much clearer when analyzing results of measurement on the PTB aluminum hole-plate [6].

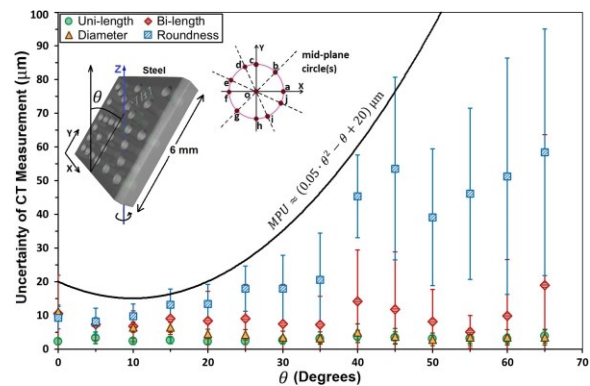


FIGURE 7. Mean uncertainties ($k = 2$) for CT dimensional measurements on the PTB steel hole-plate. Vertical bars cover two standard deviations from the mean.

To further verify the prediction trends, uncertainties for the X-ray CT dimensional measurements taken on the steel PTB hole-plate were estimated using the substitution method from ISO/TS 15530-3:2009 [16]. Individual uncertainties were computed for each measurement task [5, 15] involving the 28 holes. A total of 165 different measurements were

analyzed, 62 uni-directional (center-to-center) lengths, 47 bi-directional (point-to-point) distances, and the diameters and respective roundness measurements of the 28 reference circles at the mid-plane. Figure 7 shows the mean values (averages) of the expanded uncertainties ($k = 2$) obtained for each category with the respective quantification of data dispersion assessed by standard deviations. It is to note that this analysis was performed on unfiltered results—no smoothing filter nor outlier elimination applied for extracting dimensional data—since the averaging process already plays the role of a filter. From Figure 7 the trend is clear, the region of optimal angular orientations (θ) that minimizes the uncertainty of the overall CT dimensional measurements is close to 10 degrees, with uncertainties measured in the tens of micrometers. This coincides with the optimal region of angular orientations (θ) predicted by $D_{sim}(\theta)$ in Figure 5, and also from the previous analysis based on experimental data for measurement deviations between CT and CMM. The $MPU(\theta)$ curve shown in Figure 7 is a phenomenological approximation (based in a quadratic function) for what could be the maximum permissible uncertainty for a dimensional CT measurement task on the PTB steel hole-plate. Although measurements of diameters and uni-lengths have uncertainties under 10 μm for the full span of tilt variations, after $\theta \approx 35$ degrees, the uncertainties for measurement of bi-directional lengths and roundness measurements become very large, especially for measurements of form which can reach uncertainty values in the range from 50 to 100 μm .

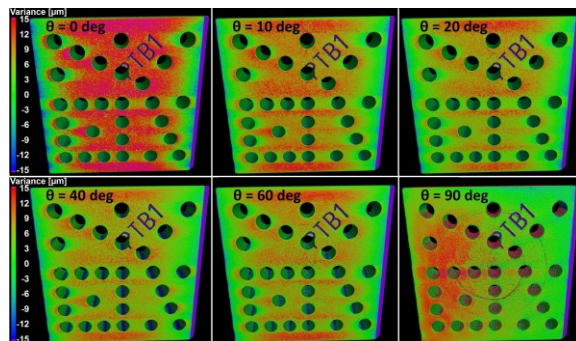


FIGURE 8. Part-to-CAD comparison for the PTB steel hole-plate CT scanned at different tilts.

The problem of finding optimal angular orientations that minimize the majority of the measurement deviations in dimensional CT is a

challenging one. It is not just a matter of looking for the best orientation in the cone-beam CT system that would avoid underrepresented areas in the Radon-space, but also minimizing beam hardening and scattering radiation. These considerations are in theory contained in the Dreamcaster simulation, see references [2, 3, 4]. However, it is important to verify that it reproduces the experimental trends. Figure 8 for example, shows a part-to-CAD comparison for the steel hole-plate at different angle orientations (θ). Surface for the part was extracted from CT scan data using the locally adaptive method in VGStudio MAX 2.2 [17]. From the comparison in Figure 8, it can be seen that deviations from the nominal geometry (CAD model) reached up to $\pm 15 \mu\text{m}$ and more according in the color-coded areas (magenta being bigger than 15 μm and violet being less than 15 μm). Some of these deviations are due to the appearance of CT artifacts—the cut corner on the right side and the (PTB1) stamp on the upper right corner of the hole-plate should not be considered deviations or CT artifacts because they were deliberately inserted as references to orient the part. At $\theta = 0$ degrees, artificial features appear on the planar faces of the hole-plate (highlighted in red in Figure 8) and most likely were produced by a combination of beam hardening and scattering effects. There are also some small depressions (blue spots) around the outer edges of the planar faces, most likely due to cone-beam artifacts (surfaces underrepresented in the Radon-space, the $P_{ct}(SURS)$ from Figure 6). By increasing the tilt angle (θ) from 0 to 10 or 20 degrees, the artifacts on the planar faces diminish, but then deviations in the order of -6 to -12 μm appear on the holes' surfaces (indicated by blue lines in Figure 8). These 'depression' artifacts seem to increase as the angular inclination θ increases. At $\theta = 40$ degrees for example, they reach deviations from the nominal geometry below -15 μm (indicated now by purple lines). According to the Dreamcaster simulation (Figure 5 and Figure 6), most of these artifacts come from X-ray beam hardening effects and some level of scattering. After $\theta = 60$ degrees, noise-like artifacts appear on top of the 'depression' artifacts that are along the holes' surfaces. These artifacts are particularly high for angular inclinations between 70 and 90 degrees, and this is why dimensional measurements with the software Calypso 5.6 were not possible at the cylinders' mid-circles, so there is no experimental data reported at those angles in Figure 5. The Dreamcaster simulation predicts high alterations in the CT data for the

region 70 to 90 degrees (see Figure 5 and Figure 6), which is coincident with the experimental observations from part-to-CAD comparison shown in Figure 8. At $\theta = 90$ degrees, the appearance of CT reconstruction artifacts is worse, see Figure 8, as large noise-like artifacts are observed in the holes' surfaces plus additional artifacts (rings) on the hole-plate flat surfaces. These artifacts are caused by a combination of large amount of beam hardening, scattering effects, and cone-beam artifacts. They were also predicted by the Dreamcaster simulation $D_{sim}(\theta)$ (see Figures 5 and 6), given the accentuated growth of $Avg(TPL)$ and $Max(TPL)$, and large percentage of surface areas underrepresented in the Radon-space ($Pct(SURS)$). From the part-to-CAD analysis above, it can be concluded that to minimize artifacts in the reconstructed CT volume for the PTB steel hole-plate, the region of optimal specimen inclination (θ) for CT scanning is in the neighborhood of 10 to 20 degrees. This is in agreement with the Dreamcaster prediction and with measurements of deviations between CT and CMM dimensional data (see Figure 5).

CONCLUSIONS

Effectively, to minimize the deviations between CT and CMM dimensional data, the optimization of X-ray CT measurements is possible by choosing appropriate angular orientation of the workpiece in the cone-beam geometry of a CT scanner. An optimal orientation leads to a reduction of uncertainties in dimensional CT measurements in the order of a few to tens of micrometers. From the analysis presented in this paper, it was shown that simulated data obtained using ray casting and Radon-space analysis with the Dreamcaster tool [2] produces trends comparable with experimental results. Predicting optimal angular orientations for CT scanning is important for optimizing CT dimensional measurements and minimizing measurement uncertainties. It is clear that, with suitably fast simulation speeds, a spectrum of orientations can be readily simulated. Using the combined factor $D_{sim}(\theta)$ as a simple estimator can be a sufficient way to readily identify optimal specimen orientations. For the workpieces examined in this work—the rectangular blocks and the PTB steel hole-plate—three basic zones of axial tilt θ (for the workpiece's main axis with respect to the CT rotating axis Z) were found: a 'green zone' from 10 to 35 degrees in which deviations between CT and CMM are minimum, a 'red zone' from 0 to 7.5 degrees and from 60 to 90 degrees in which the

largest deviations are measured, and 'yellow zones' between the two. The analysis showed that when the optimal orientations are used for scanning, the uncertainties of dimensional CT measurements can be under 10 μm . At the worst orientations, uncertainties can be several tens of micrometers, especially for measurements of form, which can reach uncertainty values ranging from 50 to 100 μm . In general, the smaller uncertainties for dimensional metrology characterization with CT are obtained for measurements of the circles' diameters and for the uni-directional (center-to-center) lengths. The bi-directional (point-to-point) lengths and measurements of form usually have larger uncertainties.

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STRAIGHTNESS MEASUREMENT BASED ON KNIFE-EDGE SENSING

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INTRODUCTION

Straightness error is a parasitic translation along a perpendicular direction to the primary displacement axis of a linear stage. Its measurement and compensation are critical in the metrology, calibration, and manufacturing of multi-axis platforms. For example, the inverted XYZ platform in Figure 1 which is a stack of three mutually orthogonal linear stages. Although the linear stages are expected to move along perfect single axis trajectories, in practice they have two lateral straightness errors as shown in Figure 2. Thus, measuring and/or calibrating the straightness errors is important in precision engineering applications including semiconductor processing, printed circuit board (PCB) processing, micromachining, precision assembly processing, *etc.* [1].

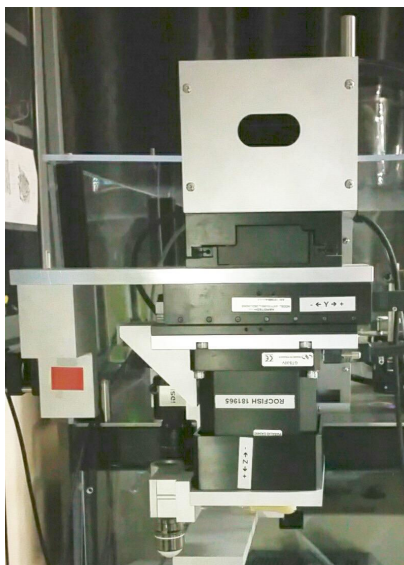


FIGURE 1. Rocfish three-axis system

Several implementations for measuring straightness error have been presented, which are

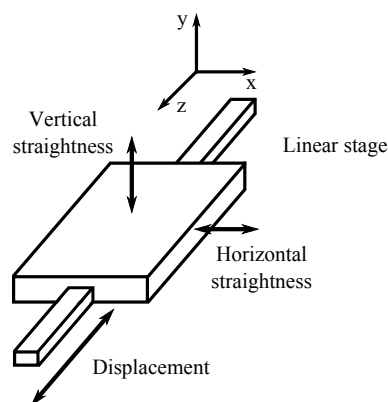


FIGURE 2. Three degrees of freedom of linear stage motion.

mostly based on three principles [2]. The first is an interferometric method, whose principle is similar to a displacement measuring interferometer. It measures the optical path change due to the movement of polarizing or non-polarizing straightness optics, and then converts the optical path change to the transverse displacement (straightness) [1, 3, 4]. The second is via light intensity detection. The straightness error produces a change of the optical power reaching a final photodetector. By monitoring and calibrating the power change, the straightness can be measured [2, 5, 6]. The third is a spot position detection method. It uses a laser as the reference location. The stage's straightness error leads to lateral motion of the laser and a final photodetector measures the position change of the incident beam [7, 8, 9, 10]. These solutions employ different configurations, components, detectors, and signal processing algorithms, and achieve varying levels of accuracy and resolution.

Our optical knife-edge method is a particularly simple configuration for light intensity detection

which builds on previous research [5, 6]. In this paper, we propose a new configuration of the knife-edge method, and explore its precision, resolution, linearity and feasibility to measure two-dimensional straightness errors.

OPERATING PRINCIPLE

The knife-edge method uses a sharp edge to block part of a laser and uses a photodetector to measure the transmitted and diffracted light. In previous studies, Fan, *et al.* presented a configuration whose experimental result showed that the accuracy is better than $\pm 0.2 \mu\text{m}$ [5]. Lee, *et al.* implemented a prototype 3-D printed stage which claims a 5.2 nm resolution with high linearity [6].

Lee's configuration (see Figure 3) splits a laser into two beams, and the two beams are partially blocked by two outward facing knife-edges. Two photodetectors measure the transmitted and diffracted light at a specified distance from the two knife-edges. When the knife-edges displace laterally, the two detected signals increase or decrease in a differential manner. Differential post-processing on two detected signals extracts the straightness error.

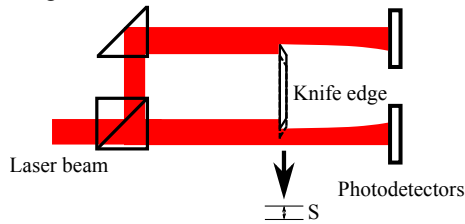


FIGURE 3. Lee's configuration based on knife-edge detection [6].

To extend the above configuration to a two-dimensional straightness measurement and reduce alignment drift errors, we propose a configuration that combines light intensity detection with spot position detection. It uses a two-dimensional knife-edge, which combines four inward facing knife-edges as a square aperture. When the single laser beam reaches the square aperture, the transmitted and diffracted light will be detected by a quadrant photodetector. When the aperture has a lateral displacement, the position of the spot on the detector will change (see Figure 4).

The quadrant photodiode consists of a 2×2 photodiode array with a small (several micrometers) gap separating adjacent elements. Due to the position of the incident beam change, the optical

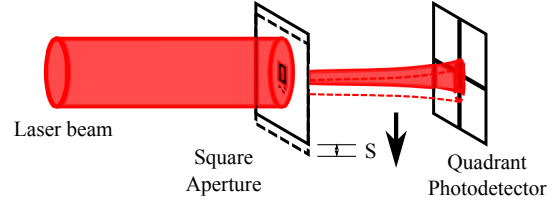


FIGURE 4. The configuration of proposed 2D straightness measurement system based on 2D knife-edge (square aperture).

power distributes differently on each element (see Figure 5). Each single photodiode converts the incident optical power to a photocurrent. By tracking these four photocurrents, the position of the spot can be derived according to [9, 11]

$$S_x = k_x \frac{(I_A + I_B) - (I_C + I_D)}{I_A + I_B + I_C + I_D} \quad \text{and} \quad (1)$$

$$S_y = k_y \frac{(I_A + I_C) - (I_B + I_D)}{I_A + I_B + I_C + I_D}, \quad (2)$$

where S_x and S_y are the straightness errors in the x- and y-directions respectively, I_A , I_B , I_C , and I_D are the photocurrents generated by the four elements, and k_x and k_y are scaling factors that convert the current measurement to its corresponding length (straightness).

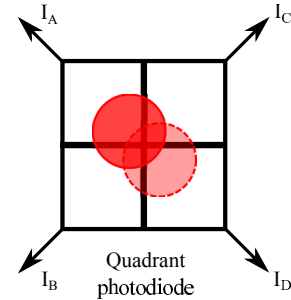


FIGURE 5. The relative position change makes the optical power distribution on the detector change, thus leading to a change in photocurrent.

The variation in the laser intensity is eliminated by dividing the sum of four elements, and the common mode noise is also eliminated by the differential post-processing (numerator part). However, the intensity distribution of the beam is not uniform because the knife edge clips the beam. It leads to the relationship in (1) and (2) which are nonlinear, thus, k_x and k_y are non-constant values, which must be calibrated carefully. The size of the aperture also leads to a trade-off between measuring range and resolution [8, 11]. However, this tradeoff can be critical in extending the range of this sensing architecture to the mesoscale, which has limited low cost, high

precision options.

IMPLEMENTATION

In this work, we utilize a one-dimensional piezo-stage with a calibrated internal capacitance sensor for closed loop control to simulate a horizontal straightness error. Thus, we designed a configuration to verify the one-dimensional measuring capability of the proposed sensor only. Figure 6 shows the configuration to measure one-dimensional straightness error. Unlike Lee's configuration, it uses a single beam, two inward-facing knife-edges, and a quadrant photodetector. This makes the configuration more compact and easier to align.

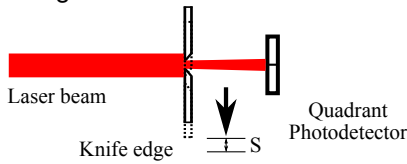


FIGURE 6. The proposed configuration to measure one-dimensional straightness error.

SIMULATIONS

Simulations were performed to determine the theoretical relationship between the straightness and photocurrents generated by the detector. When the laser reaches the aperture, part of the light passes the aperture, and part of the light is blocked. Due to diffraction, some transmitted light "bends" into the region of the geometrical shadow of the aperture. The size of the spot on the detector is not exactly the same as the aperture. To obtain the relationship between lateral displacement and the photocurrents, the distribution of the transmitted light must be identified.

To identify the distribution of the transmitted light on the detector, we use angular spectrum propagation to calculate the spot distribution on the detector plane at distance z [12],

$$U(x, y, z) = F_{f_x f_y}^{-1} (F_{f_x f_y} (U(x, y, 0)) \cdot e^{j \frac{2\pi}{\lambda} \sqrt{1 - \lambda^2 f_x^2 - \lambda^2 f_y^2} z}), \quad (3)$$

where $U(x, y, 0)$ is the distribution across the aperture plane at distance $z = 0$, $U(x, y, z)$ is the distribution across the detector plane, $F_{f_x f_y}$ and $F_{f_x f_y}^{-1}$ are the two-dimensional Fourier transform and its inverse transform, and f_x and f_y are the spatial frequency variables.

After obtaining the complex field $U(x, y, z)$, the intensity at the detector plane is calculated as the

modulus squared,

$$I(x, y, z) = |U(x, y, z)|^2. \quad (4)$$

The intensity pattern is detected by the photodetector, which converts the optical power to photocurrent. When using a quadrant photodetector, the photocurrent generated by each element is related to the integration of the intensity pattern on that element,

$$I_i = \int_{A=i} R_\lambda I(x, y) dA, \quad (5)$$

where I_i is the photocurrent generated by the i th element, R_λ is the responsivity or photosensitivity of the photodiode, measuring the effectiveness of the conversion of optical power into photocurrent, expressed in units *amps/watt*.

Figure 7 shows the simulated power distribution of the original beam and the power distribution on the photodetector after the beam passes the two knife-edges (MATLAB simulation). The knife-edges reshape the beam through diffraction.

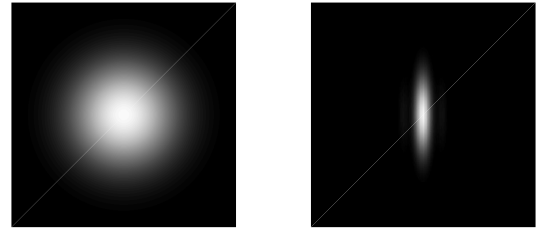


FIGURE 7. The left figure simulates a Gaussian laser with a 2.6 mm beam diameter, and the right figure simulates the laser beam after passing a 0.2 mm-width slit and being received by a detector 40 mm away.

As the relative positions of the knife-edge and detector change, the quadrant photodetector generates varied photocurrents. Figure 8 shows the relationship between straightness error and photocurrents. The middle linear part is an active area, where differential processing (1) can be applied to determine the one-dimensional straightness error.

EXPERIMENTS

Experiments were performed to compare the measurement accuracy and resolution with and without the knife-edge, and explore whether the setup with knife-edges is superior to that without knife-edges. Figure 9 shows the setup of the measurement system with knife-edges. The green frame holds the two razor blades

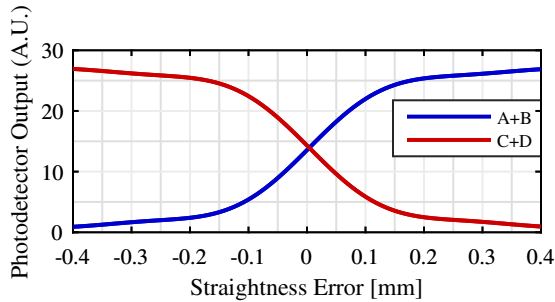


FIGURE 8. The photocurrents generated by the left two elements and right two elements change in opposite directions when the aperture and detector have a relative lateral displacement.

as two inward-facing knife-edges. A quadrant photodetector is placed on a piezo-stage, which produces a lateral displacement imitating the straightness error. The alternative setup removes the knife-edge, so that the beam is directly incident on the quadrant photodetector.

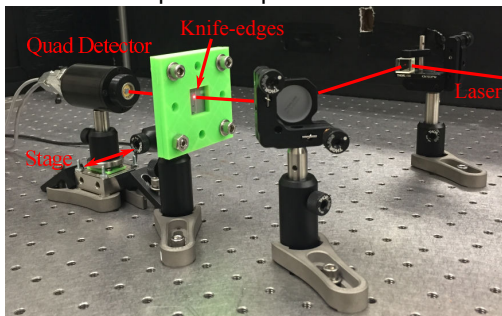


FIGURE 9. The setup of the 1D straightness error measurement system with two inward-facing knife-edges.

As previously mentioned, the scaling factor k_x in (1) is not a constant value because the relationship between the left and right terms is nonlinear. Thus, before using this system to measure straightness error, it must be carefully calibrated. There is an embedded capacitive sensor in the piezo-stage, whose output can be used as the reference. We drive the piezo-stage to sweep in a sawtooth path within $\pm 40 \mu\text{m}$, and then curve fit the photodetector's readout to the reference. In this way, the scaling factor k_x can be determined. Figure 10 shows the relationship between the knife-edge system readout and the reference. The slope of the curve is the scaling factor k_x .

After obtaining the scaling factor k_x as a function of knife-edge system readout, the measurement result can be derived. We test both setups

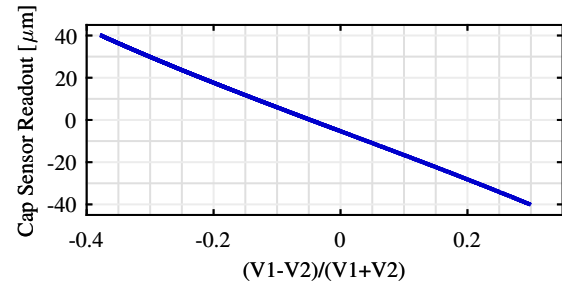


FIGURE 10. The proposed measurement system readout with respect to the embedded capacitive sensor readout. The stage is driven to linearly move within $\pm 40 \mu\text{m}$.

with and without knife-edges. Figure 11 shows the comparison. From the figure, we can see the setup with knife-edges can achieve a 10X improvement in sensitivity and noise than that without knife-edges. To generate the figure, the piezo-stage was driven in a sawtooth path within $\pm 40 \mu\text{m}$.

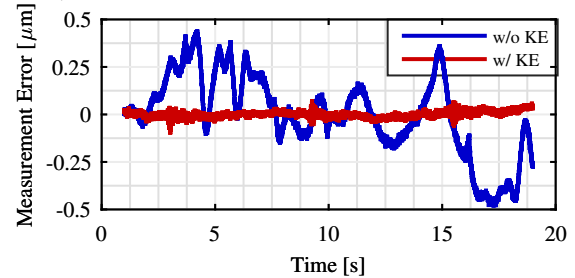


FIGURE 11. The red curve is from a measurement using knife-edges, whose measurement error is within $\pm 40 \text{ nm}$. The blue curve is from a measurement setup without knife-edges, whose measurement error is within $\pm 400 \text{ nm}$.

The knife-edges block a portion of the beam and narrow the spot size, which increases the measurement sensitivity, but decreases the measurement range. The accuracy of this method relies on the power stabilization of the laser source. Experiments have observed small intensity drifts of the incident beam. Therefore, a robust system will need to compensate the drift through better intensity stabilization, laser power monitoring, or AC modulation of the incoming light.

CONCLUSION

We have presented a simple and compact configuration to measure the straightness error of a linear stage which uses inward-facing knife-edges, a single laser beam, and a single quadrant detector. Currently, we have tested its performance to measure one-dimensional

straightness error, and it can achieve an accuracy of ± 40 nm. It is also feasible to extend this sensor to a two-dimensional straightness error measurement setup.

FUTURE WORK

It has been shown that the inward-facing knife-edge configuration is feasible to measure a one-dimensional straightness error with a ± 40 nm noise level. In the future, we will extend this configuration to a two-dimensional straightness measurement (Figure 4). To minimize the effect of laser power instability, we plan to monitor the laser power during the measurement, and compensate the measured signal in processing. Lastly, we will investigate the theory of the aperture design and how it affects range versus sensitivity tradeoffs.

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LOW-COST AUTOMATED WAFER HANDLING SYSTEM FOR HIGH THROUGHPUT WAFER METROLOGY

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ABSTRACT

This paper introduces the design of a robotic system that enables automated wafer handling to support inline wafer metrology in semiconductor manufacturing operations. The device has three degrees of freedom configured to enable motion of an end effector across a cylindrical work space. The end effector is able to grip wafers from a cassette, translate them across space, and position them into a passive alignment mechanism. Combining the wafer handling system with a passive alignment stage provides a low-cost and high speed method for obtaining micron level positioning repeatability. Translational repeatability better than 50 μ m and rotational repeatability better than 0.15° were demonstrated with a wafer insertion time on the order of 15 seconds.

INTRODUCTION

High-throughput wafer scale metrology is a critical component of the nano and microscale manufacturing processes. Inspecting nanoscale features over large areas is typically only possible off-line due to the time required. Recent studies by Duenner [1] and Yao [2] demonstrate nanoscale wafer metrology at speeds of up to 45 wafers per hour (WPH). The system consists of a passive wafer alignment stage kinematically coupled to an atomic force microscope (AFM) scanning stage. One of the largest components of time spent per wafer of their metrology system is the manual insertion and removal of wafers into the wafer stage.

The wafer handling robot presented in this paper increases the throughput and accuracy of the passive wafer alignment system described by Duenner et al. [1]. The combined system allows for high throughput wafer scale metrology with nanoscale resolution at a cost several orders of

magnitude lower than that of competing systems.

DESIGN

The automated wafer handling system operates in a cylindrical workspace with one rotational axis and two translational axes. Figure 1 shows a CAD rendering of the system. The cylindrical coordinate system was chosen to minimize the footprint of the wafer handling robot as well as the number of actuated degrees of freedom. The serial configuration of the axes requires that the axis connected to ground have relatively low errors as they will propagate to other axes. A comparison of motion stages on the basis of cost and precision demonstrated that rotational stages can be manufactured for low cost and maintain high precision using significantly fewer parts than a translational stage. For this reason, a rotational stage was chosen for the axes connected to ground.

The rotational axis consists of two pulleys connected by a toothed belt. This method of rotational power transmission was chosen for several reasons. When compared to a gear-driven system, toothed belts allow relaxed machining tolerances and overall lower cost. Furthermore, toothed belts exhibit low backlash when properly tensioned. The pulleys are sized to have a reduction ratio of 4:1 with the driven pulley being the smaller of the two. The larger pulley is connected to the rotating base of the wafer handling system. A low-cost “lazy-susan” turntable was utilized in place of a more conventional ball bearing to save cost. The turntable consists of an inner and outer raceway with spherical steel balls separating the two cast aluminum raceways. The balls in the turntable provide low resistance to rotation of the stage and the turntable is rated to support substantial axial loads.

Two translational axes are stacked in series on top of a platform fixed to the outer raceway of the turntable bearing. One axis translates radially with respect to the rotating axis and the next axis (attached to the radial axis) translates in an axial direction with respect to the rotating axis. Each translational axis consists of a carriage with four wheels sliding on an extruded aluminum rail. Eccentric nuts on two of the wheels allow for the application of a preload to increase out of plane stiffness of the linear axes. The carriages are connected to lead screws by anti-backlash nuts that help improve repeatability of linear motion. All axes are driven by 1.8 deg/step stepper motors which allow for sub-mm resolution motion without the need for closed loop controls.

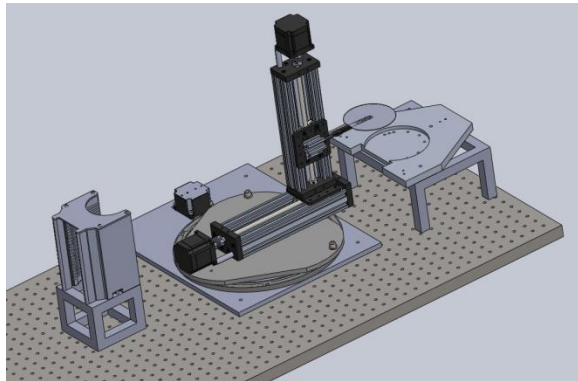


FIGURE 1: CAD RENDERING OF WAFER HANDLING MACHINE

A vacuum wand was utilized as an end effector to engage and release wafers from the wafer handling robot. A circuit was constructed to allow actuation of the vacuum wand via a digital output port on the motor driver.

The system is designed to transport wafers from a standard wafer cassette to a passive wafer alignment stage described by Duenner in [1]. The passive alignment stage is enhanced with the addition of an automated nesting force actuator. A voice coil actuator is rigidly attached to a flexural member that restricts the motion of the actuator to be normal to the circumference of the wafer. A custom transconductance amplifier circuit is used to convert a voltage signal from a DAQ card to current through the voice coil amplifier. Current through the amplifier is linearly

proportional to force generated by the amplifier, and for low operating frequencies a linear relationship is maintained between voltage output from the DAQ and force applied to the wafer. This results in an easy to control and highly repeatable method for applying a nesting force to the passive wafer alignment system. The complete prototype system is shown in Figure 2.

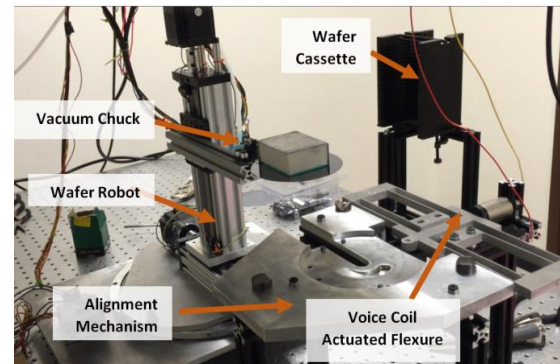


FIGURE 2: PROTOTYPE WAFER HANDLING SYSTEM

CONTROLS

Three stepper motors actuate the robot with three degrees of freedom. The stepper motors are controlled by a TinyG [3] motor driver which accepts plain-text machine code from a computer sent via serial protocol. A LabVIEW virtual instrument was developed to provide G-Code commands to the stepper motors in coordination with application of the nesting force and vacuum from the vacuum wand.

ERROR BUDGET

An error budget was created to predict the overall system performance before actually building the prototype. Tests of repeatability for both the linear stages and the rotary table were carried out to determine the individual impact of each stage. Repeatability measurements for the rotary stage consisted of collecting data over 50 trials, where each trial consisted of rotating the stage 180°, and then -180° back to its starting position, where the deviation from its original position was then measured. Deviations from the original position were measured with a fixture having six capacitance probes from Lion Precision. The probes have a range of 250

microns and a resolution of 10 nanometers [4]. The method for calculating translational/rotational deviations and repeatability in six degrees of freedoms is discussed the “Testing and Results” section of this paper. Average translational repeatability of 25.5 microns, and an average rotational repeatability of 0.014 degrees were recorded for the rotational stage. Individual tests of positioning repeatability were carried out in a similar fashion for both of the linear stages. Average translational repeatability of 11.1 microns and rotational repeatability of 0.11 degrees were recorded for the translational stages.

A series of heterogeneous transformation matrices (HTMs) were utilized to represent the propagation of repeatability error from the individual stages to error between the end effector and wafer. In this study error terms were the 1- σ repeatability values for each actuated axis. Figure 3 shows the local coordinate frames utilized to create the error map for the system. After determining the transformation from tool tip to reference coordinate frame, transformation from the work surface (wafer) to the reference frame was calculated. The relative error in the reference coordinate system was calculated by taking the cross product of the inverse of the tool tip HTM matrix and the workpiece HTM matrix [5]. Estimated position error from our error budget is shown in Table 1 and was calculated using an error budget spreadsheet described by Slocum in [6].

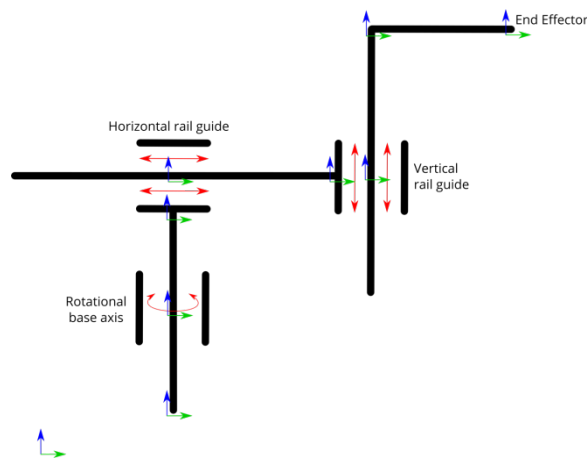


FIGURE 3: RELEVANT COORDINATE REFERENCE FRAMES

TESTING AND RESULTS

Repeatability for the combined wafer loading robot was performed as follows. The two linear axes were extended to their maximum displacement. The base was then rotated 180 degrees while at the same time the linear axes returned to their home position. The base was then rotated 180 degrees back to its original position, while simultaneously extending its linear axes out to place the end effector back to its original starting point. Displacement measurements were once again taken with capacitance probes at the starting location, with 50 trials taken for this experiment. The travel moves chosen for these measurements were intended to mimic the movements that would be made for an actual wafer loading operation.

All repeatability measurements were taken with reference to the metrology frame shown in Figure 4. Linear displacements of the capacitance probes were recorded relative to an initial measurement. Matrix inversion was utilized to transform linear measurements from the capacitance probe fixture into three translational and three rotational error measurements. Rotational errors were assumed to be small so small angle approximations were used and second-order angular errors ignored. The transformation shown in Equation 1 is a function of the dimensions l_1 through l_4 shown in Figure 4.

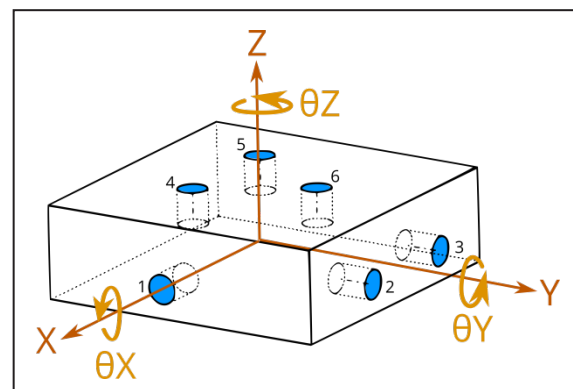


FIGURE 4: SIX DEGREE OF FREEDOM CAPACITANCE PROBE FIXTURE

EQUATION 1: DISPLACEMENT TO ERROR TRANSFORMATION MATRIX

$$\begin{bmatrix} T_x \\ T_y \\ T_z \\ R_x \\ R_y \\ R_z \end{bmatrix} = \begin{bmatrix} -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 0 & 0 & \frac{l_1}{2} \\ 0 & -1 & 0 & 0 & 0 & -\frac{l_1}{2} \\ 0 & 0 & -1 & l_3 & \frac{l_2}{2} & 0 \\ 0 & 0 & -1 & l_3 & -\frac{l_2}{2} & 0 \\ 0 & 0 & -1 & -l_4 & 0 & 0 \end{bmatrix}^{-1} \begin{bmatrix} \delta_1 \\ \delta_2 \\ \delta_3 \\ \delta_4 \\ \delta_5 \\ \delta_6 \end{bmatrix}$$

After transforming displacements measured by the capacitance probes into translational and rotational errors, the RMS value was computed for each of the translational and rotational error components for each trial. The results are shown in Figure 5 and Figure 6. The standard deviation of these errors relative to the initial measurement are shown in Table 1 compared to the error values predicted from the error budget.

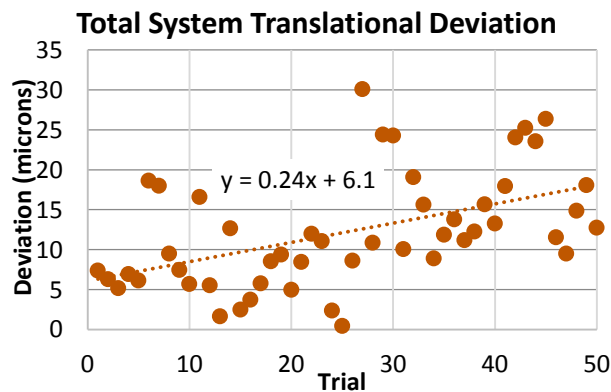


FIGURE 5: TRANSLATIONAL REPEATABILITY

The error test indicated that the system is capable of adequate performance to be adapted in the high-throughput metrology system presented by Yao [2]. By proper parameters input into the control unit, the wafer handling system is able to secure the wafer into the passive alignment stage automatically and reduce the time budget of overall inspection procedure. An insertion time of 15 seconds is reached. Moreover, introducing this handling system also significantly removes the chance to accidentally damage the wafer by human operation. These advantages make an in-line

inspection of wafers with nano-scaled patterns be possible.

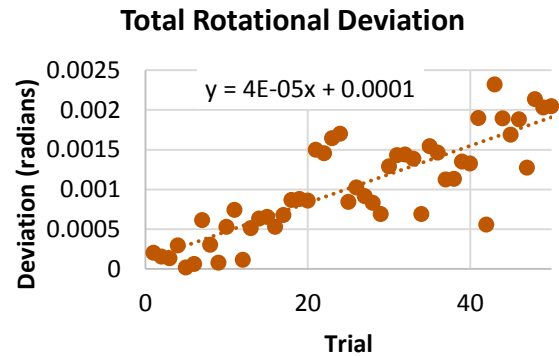


FIGURE 6: TOTAL ROTATIONAL REPEATABILITY

TABLE 1: COMBINED RANDOM ERROR PREDICTED VS. MEASURED

	Translation (microns)			Rotation (milliradians)		
	X	Y	Z	θx	θy	θz
Predicted	107	35	85	.35	.11	.35
Actual	10	5	5	.139	.47	.52

After combining this wafer handling system with the in-line inspection system presented in [2], the combined system still requires a manual operation to remove the top scanning stage for wafer loading into the passive alignment stage. In order achieve a totally automatic system, it is necessary to build a mechanism to automate the lifting of the scanning stage. With the addition of an automated stage placement system, the in-line inspection system has a high potential to be integrated into nanomanufacturing in order to assist with quality control and increasing yield.

CONCLUSION

The wafer handling system presented in this paper increases the utility of high-throughput metrology system presented by Yao [2] and Duenner [1] while increasing its throughput and accuracy. The combined system allows for automated in-line inspection of wafers with nanoscale resolution at a cost two orders of

magnitude less than that of conventional inline wafer inspection systems.

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ECDM using micro/nanoelectrode fabricated by focused-ion-beam chemical vapor deposition

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INSTRUCTIONS

Micro electro-discharge machining (micro-EDM) and micro electro-chemical discharge machining (micro-ECDM) play important roles in the precision engineering, especially for manufacturing dies and other difficult-to-machine materials. Preparation method of the tool electrode, attracts the interests of many researchers. In 1980s, Masuzawa *et al* proposed the method named as wire electro-discharge grinding (WEDG), in which the tool electrode is EDMed as the workpiece, using a travelling wire as the electrode along the wire guide [1]. Takahata *et al* utilized LIGA process to fabricate electrode arrays with a high aspect ratio, hence batch micro-EDM could be achieved [2, 3]. Shibayama and Kunieda developed tool electrode with microholes inside which are designed for the flow of dielectric liquid, and the electrode was applied to sinking EDM [4]. Egashira *et al* used a silicon probe which is originally used for scanning probe microscopes (SPM) as the electrode for performing micro-EDM [5]. Li and Zhao *et al* proposed bunched-electrode EDM using powerful multi-hole inner flushing, aiming at improving the efficiency of material removal [6]. Xu and Wu *et al* proposed micro-double-staged laminated object manufacturing (micro-DLOM) to fabricate 3D micro-mould, in which steel foils were firstly cut by femtosecond laser to obtain single-layer graphics and then welded using the method of micro-electric resistance slip welding [7, 8]. However, few of the methods is able to fabricate an electrode with a characteristic dimension down to submicron scale, which hinders from performing EDM or ECDM at a smaller scale and investigating the phenomenon that occurs between the micro/nanoelectrode and the workpiece.

In the field of micro/nanofabrication, focused-ion-beam chemical vapor deposition (FIB-CVD) was proposed and has been repeatedly demonstrated to be an effective tool in

fabricating various 3D nanostructures for micro/nanoelectromechanical devices (MEMS/NEMS) [9, 10]. The basic principle is, under the irradiation of energetic FIB, molecules of the source gas that surround the irradiated area deposit to solid structure. It keeps grow if the irradiation of FIB lasts. By controlling the scanning of FIB via a computer pattern generator (CPG), complex 3D structure with desired shape and sizes can be achieved [11, 12]. Structure with a characteristic dimension as small as 100 nm can be created. Structures basing on various elements such as platinum (Pt), tungsten (W) and carbon (C), so that structures with different physical and mechanical properties can be obtained by using a corresponding compound as the source gas, [13].

In this paper, FIB-CVD was utilized to fabricate micro/nanoelectrode and applied to micro-ECDM, aiming at establishing a novel effective fabrication technique of the tool electrode and investigating its fabrication characteristics.

Fabrication of micro/nanoelectrode using FIB-CVD

The micro/nanoelectrodes were fabricated using FIB-CVD via a commercially available FIB system (FEI Corp.; 200) that equips a Ga⁺ ion beam with an acceleration voltage of 30 kV. Beam current in a broad range from picoampere (pA) to nanoampere (nA) order can be chosen to achieve different fabrication efficiency and resolution. In this work, micro/nanoelectrode in amorphous Pt-based compound was deposited.

Normally, FIB-CVD is often used to deposit micro/nanostructures onto a silicon substrate, due to its various applications related to semiconductor industry and microelectronics. However in this work, aiming at using the micro/nanostructure as the electrode, the deposition should be performed onto a metallic substrate, such as copper (Cu) or stainless steel.

Moreover, the micro/nanoelectrode on the substrate is tiny comparing to the size of the substrate, inclination of a small angle between the substrate surface and the workpiece surface may leads to the short between the electrode and the workpiece, therefore, a commercially available needle in the material of stainless steel was cut and the micro/nanoelectrode was fabricated onto the tip of the needle.

Figure 1 shows the scanning electron microscope (SEM) image of a cylinder-shaped microelectrode with a diameter of 10 μm and a height of 28 μm that was fabricated on the tip surface of needle. The beam current of FIB and the fabrication time were 3977 pA and 40 minutes 44 seconds, respectively.

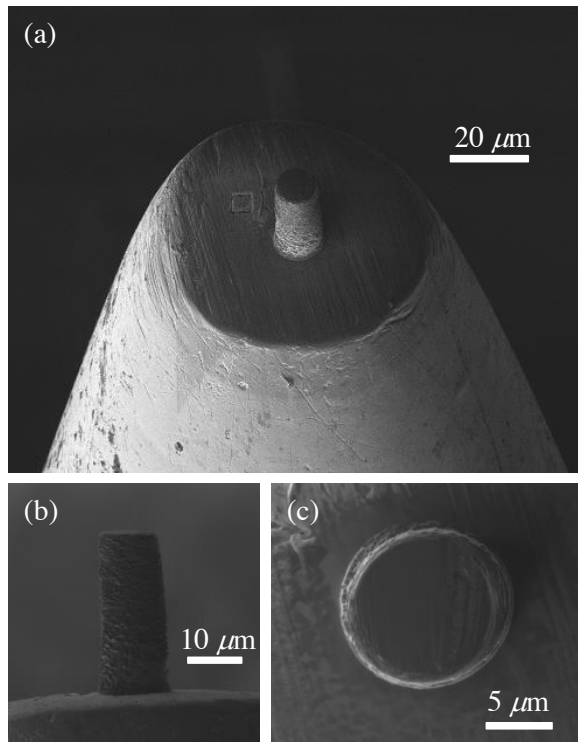


FIGURE 1. Microelectrode fabricated on the tip surface of a needle, (a) 3D, (b) side and (c) top view.

Application of microelectrode to micro-ECDM

Concerning the small electrode size and the comparatively high-rate wear of the electrode in micro-EDM, in this work, the microelectrodes fabricated by FIB-CVD were chosen to be applied to micro-ECDM. The micro-ECDM was performed with a specially-made electrochemical micro/nanomachining system, as shown in Fig. 2.

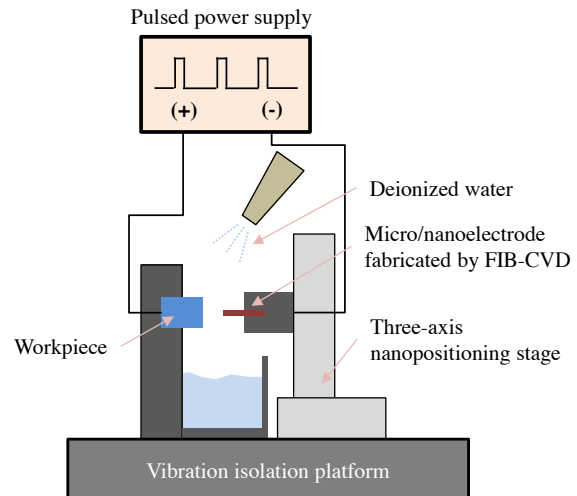


FIGURE 2. Experimental setup of micro-ECDM using microelectrode fabricated by FIB-CVD.

Key parts of the system include:

- A three-axis nanopositioning stage with a minimum pitch of 2 nm (Physik Instrumente GmbH; P-563.3CD); feed of the electrode was controlled by a unique program created using Visual C++ language on the computer, while the workpiece was mechanically fixed to be static.
- A specially-made pulsed power supply; high-frequency impulsing power source basing on FPGA was developed, with an adjustable pulse duration range of 20 ns -102 μs .

Three microelectrodes were fabricated using the same fabrication parameters as aforementioned to investigate the fabrication characteristics of micro-ECDM. Due to the small diameter of the electrodes, low open voltage 10, 12.5 and 15 V were chosen to perform micro-ECDM, respectively. Table 1 shows the machining parameters.

Three microholes were respectively machined on the surface of 304 austenitic stainless steel. Figure 3 shows the SEM image of a tilted micro-hole, which was fabricated by using the open voltage of 15 V. Hole diameters, and unilateral discharge gaps which were calculated by dividing the difference of the hole diameter and the electrode diameter by 2, are plotted in Fig. 4. With the increase of open voltage, the hole diameter and unilateral discharge gap almost linearly increases. In case of using the open voltage 10 V, the unilateral discharge gap was as small as 6.6 μm .

TABLE 1. Machining parameters of micro-ECDM using microelectrodes fabricated by FIB-CVD.

Parameters	Value	Unit
Microelectrode diameter	10	μm
Microelectrode height	28	μm
Working fluid	Deionized Water	-
Workpiece	Polished 304 austenitic stainless steel	-
Open voltage	10, 12.5, 15	V
Pulse duration	800	ns
Pulse separation	4200	ns
Working depth	20	μm
Working step	0.5	μm

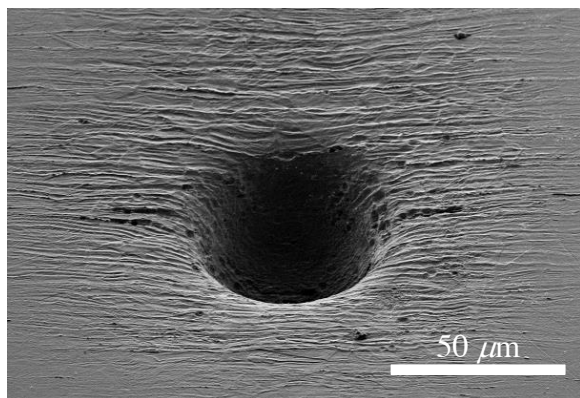


FIGURE 3. Microhole machined on the polished surface of 304 austenitic stainless steel using the open voltage of 15 V.

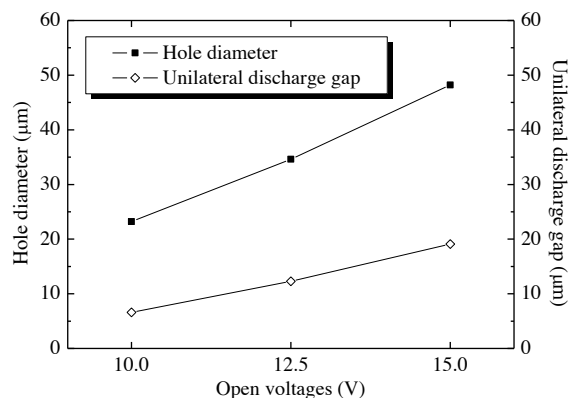


FIGURE 4. Open voltage dependencies of the hole diameter and the unilateral discharge gap.

Fabrications of typical 3D nanoelectrodes using FIB-CVD

By introducing FIB-CVD to the fabrication of micro/nanoelectrode for ECMD/EDM,

- Electrodes with a characteristic dimension at micrometer scale or submicron scale can be easily fabricated due to the high fabrication resolution of FIB-CVD. Therefore, a tool for the research on the ECMD/EDM phenomenon at submicron scale is realized.
- 3D micro/nanoelectrode with high degree of freedom can be fabricated at one-time and on one work station, hence ECMD/EDM with a higher degree of freedom than the present ones may can be realized. Unique control program of FIB [11, 12] was developed using Visual C++ language for processing the data that was designed by common 3D-CAD softwares, such as Solidworks and Vectorworks. In other words, one-click fabrication of 3D micro/nanoelectrode can be achieved via FIB-CVD.

In order to demonstrate the effectiveness of micro/nanoelectrodes fabricated by FIB-CVD, various typical 3D nanoelectrodes were fabricated on copper substrate, as shown in Fig. 5. Figure 5(a) shows a pillar type nanoelectrode with a diameter of 210 nm and a height of 6.45 μm . Figure 5(b), (c) and (d) show nanoelectrodes with a horn, a corner and an array type, for which the fabrication times are 9 minutes 51 seconds, 4 minutes 59 seconds and 1 hour 13 minutes 20 seconds, respectively. These results indicate the advantages of

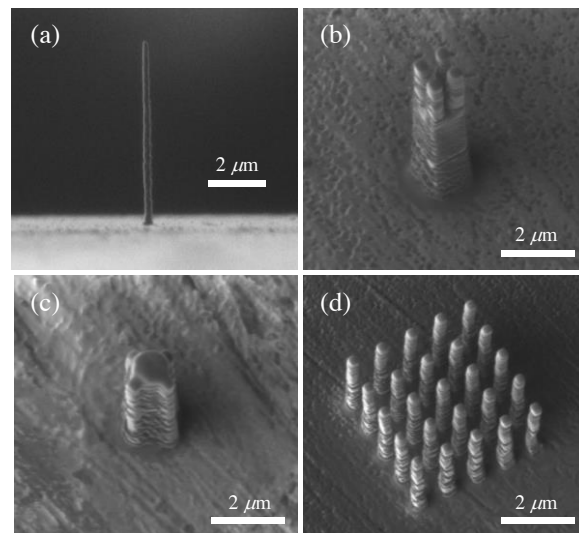


FIGURE 5. Various typical nanoelectrodes fabricated by FIB-CVD, (a) pillar, (b) horn, (c) corner and (d) array type.

fabricating micro/nanoelectrode by using FIB-CVD.

Conclusions

In this work, FIB-CVD which is an effective tool in fabricating 3D micro/nanostructure, was first time introduced to fabricate micro/nanoelectrode for micro-ECDM, with the objectives of proposing a fabrication technology of the electrode with a submicron characteristic dimension, investigating its fabrication characteristics and supplying a tool for the exploration of the ECDM/EDM science into the submicron scale. Three microelectrodes with the same diameter 10 μm and height 28 μm were fabricated on the polished tip surfaces of stainless needles. The microelectrodes were respectively applied to micro-ECDM using different low open voltages 10, 12.5 and 15 V. We found that the hole diameter and the unilateral discharge gap almost linearly increased with the increase of the open voltage. The unilateral discharge gap was found to be 6.6 μm when the open voltage was 10 V. Furthermore, various typical 3D nanoelectrodes including pillar, horn, corner, array types, were fabricated to demonstrate the effectiveness and advantages of fabricating micro/nanoelectrode using FIB-CVD.

Acknowledgements

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Design and Fabrication of an Ultrasonic Waveguide for Microchip Cooling Applications

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1. INTRODUCTION

Ultrasound has been widely used in industries such as cleaning, medical and manufacturing areas. One of challenging application area of the ultrasound is microchip cooling [1-3]. Previous researches were investigated, and main focuses of those studies were about exploring the methods of increasing passive cooling effect by enhancing the convective heat transfer coefficient. In addition, several researches were reported about thermoacoustic refrigerator system, which uses thermal energy to create acoustic waves [4-8].

But in this research, an ultrasonic waveguide for microchip cooling that is operated by 40 kHz operating frequency, was designed and fabricated for active cooling. To develop the waveguide, finite element analysis using ANSYS software was processed. First, anti-resonance frequency for a piezoelectric actuator was predicted, the result was compared with the experimental result. Then the waveguide was fabricated, and impedance characteristics were measured and compared. Next, predicted anti-

resonance frequency for the ultrasonic waveguide was compared with the experimental value. Finally, the application area of the developed ultrasonic waveguide to microchip cooling is discussed.

2. DESIGN AND FABRICATION OF THE ULTRASONIC WAVEGUIDE

For the generation of the ultrasound, the 40 kHz ultrasonic waveguide for microchip cooling is mainly composed of two parts, the ultrasonic unit and the electric generator. The fabricated ultrasonic waveguide system is shown in Figs. 1 and 2. The waveguide has a cylindrically shaped aluminium (Al) waveguide with a lead zirconate titanate (PZT) actuator attached on the top. In the waveguide, there are a resonator and stacks that can refrigerate. For the actuation, power is supplied to the PZT actuator, and the actuator vibrates. Subsequently, the displacement is transferred through the waveguide to the resonator part for cooling. The fabricated stacks are shown in Fig. 3.

To design the ultrasonic waveguide, FEM analysis using Ansys (Commercial FEM

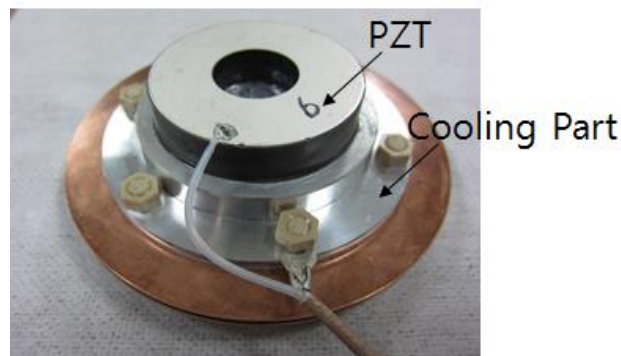


Fig. 1. Ultrasonic waveguide for microchip cooling.

software) was performed. Firstly, the PZT actuator was modeled with the analysis tool. For the analysis, it was modelled axis-symmetrically and the nodes of the top and bottom electrodes were coupled to apply voltages. To obtain a solution, series of calculations were performed from 30.0 kHz through 40.0 kHz. As a result, the highest impedance value was 34.8 kHz, which was decided as a design frequency.

Secondly, the waveguide with the PZT actuator was modelled. After analysis, the highest impedance value was 39.2 kHz as shown in Fig. 4.

For the prediction of the displacement, analysis was performed using the same analysis model. The PZT actuator and the waveguide were modeled. And modal analysis was performed and structural motion could be predicted. As a result, displacements at the operating frequency of 39.2 kHz is shown in Fig. 5.

3. EXPERIMENTS

Based on the analysis results, ultrasonic waveguide was fabricated. And the peak frequency value of impedance was measured to be 34.7 kHz of the PZT alone and 39.8 kHz of

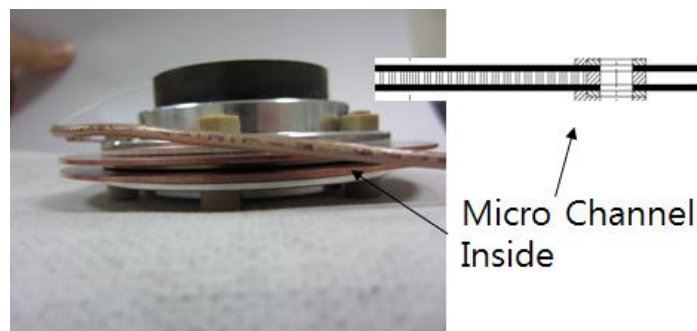


Fig. 2. Ultrasonic waveguide structure.

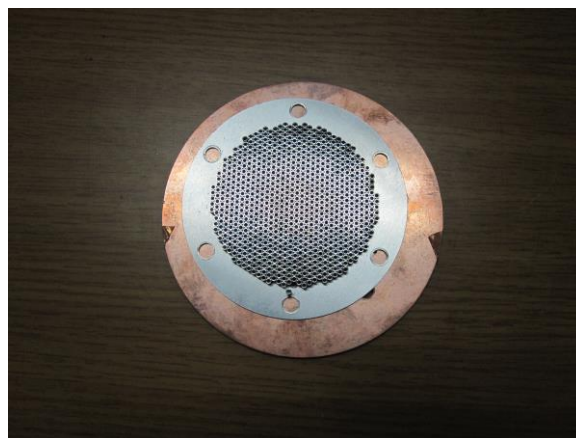


Fig. 3. Fabricated stacks.

the system, which agreed well with predicted value with 0.3% and 1.5% error, as shown in Fig. 6 for the waveguide.

4. CONCLUSION

In this paper, the design and fabrication processes of an ultrasonic waveguide for

microchip cooling that is operated by 40 kHz operating frequency were explained. In the development stage, finite element analysis using ANSYS software was performed to design the waveguide. So the predicted anti-resonance frequency for a piezoelectric actuator was 34.8 kHz, which was in good agreement with the

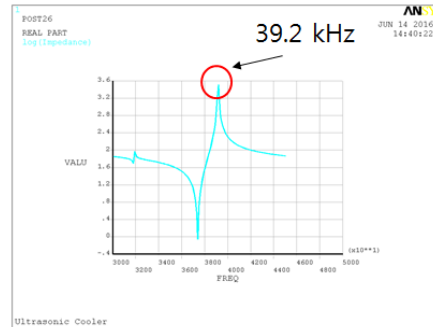


Fig. 4. FEM analysis result of the impedance graph.

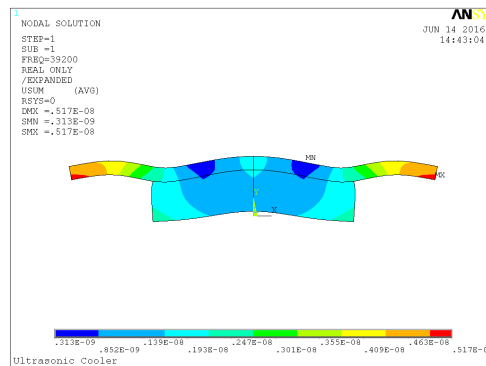


Fig. 5. FEM displacement analysis result.

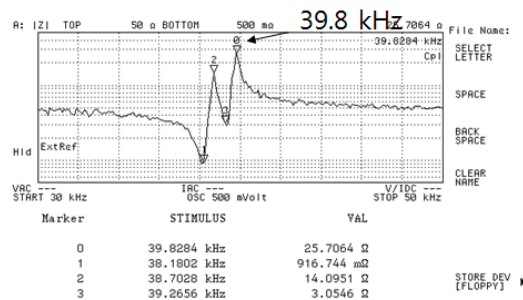


Fig. 6. Experimental results of the impedance graph.

experimental result of 34.7 kHz with 0.3% error. The waveguide was manufactured, and impedance characteristics were measured and compared. The predicted anti-resonance frequency for the ultrasonic waveguide was 39.2 kHz, which coincided with the experimental value of 39.8 with 1.5% error. Based on these results, it is thought that the developed ultrasonic waveguide will be applicable in microchip cooling.

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HARD X-RAY MICROSCOPY WITH SUB-20 NM SPATIAL RESOLUTION: INSTRUMENTATIONAL CHALLENGES AND APPROACHES

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INTRODUCTION

X-ray microscopy is a powerful tool routinely used for characterizing and understanding of internal structure and dynamic processes in various material systems. Imaging resolution is determined by the size of the focused beam delivered by the x-ray microscope. [1–3] Different types of x-ray focusing optics, (which include Kirkpatrick-Baez elliptical mirrors, Fresnel x-ray zone plates (ZP), kinoform, wave guides, refractive x-ray lenses) can be used to achieve nanometer scale spot size. [4–7] Multilayer Laue Lenses (MLL) were proposed as an alternative approach to achieve nanofocusing and overcome well-known ‘aspect ratio problem’ of the zone plates for the hard x-ray regime. [8] Individual MLL can be considered as a 1D zone plate with an additional requirement to satisfy Bragg condition in order to achieve optimal performance. In the MLL microscope, a pair of MLLs needs to be aligned orthogonally with respect to each other as well as to satisfy Bragg conditions to achieve point focusing, as shown in Fig. 1(a). In practice, this requires an instrument capable of delivering over twenty degrees of motion with very high stability and low thermal drifts. In this work, we present the first MLL-based X-ray microscope developed at the National Synchrotron Light Source II, Brookhaven National Laboratory, which offers 11x13 nm² spatial resolutions x-ray imaging. We then introduce recently developed 2D MLL optics suitable for the next generation of scanning X-ray microscopes.

DESIGN OF THE MLL MICROSCOPE

In order to achieve nm-scale spatial resolution and minimize long-term thermal drifts, the microscope should have small form factor, low heat dissipation and stiff actuators. Moreover, developed microscope should be vacuum compatible to enable measurements at

cryogenic temperatures. Figs. 1(b) and 1(c) show the microscope we constructed. We built the MLL microscope utilizing high stiffness piezo actuators operated through a stick-slip principle. Schematic of the microscope along with a computer-aided design (CAD) model and photographs of the actual system are shown in Fig. 1(b-d), all major components are enumerated. The microscope comprises of different functional modules, in which 1 is the vertical MLL assembly; 2 is the horizontal MLL assembly; 3 is the Order Sorting Aperture (OSA) assembly; 4 is the sample stage assembly. Each of the modules is equipped with different types of standard and customized piezo stages manufactured by Attocube AG, Smaract GmbH and PI/Micos GmbH. In total, the microscope consists of 20 various piezo-motors. During the x-ray experiments, a sample is raster scanned with respect to the incoming x-ray beam. For fine sample scanning, we utilize piezo scanners with built-in flexure amplifiers, mounted on top of the coarse piezo motors, to enable imaging in the 50x50x50 μm^3 volume. To track position of the individual components and ensure adequate compensation of all thermal drifts, a total of nine fiber-optic interferometers monitor the X, Y and Z positions of the vertical, horizontal MLLs and the sample assemblies. Mounted on a common Invar reference frame, all interferometer heads (AttoFPSensors; see ref. [9] for characterization details) monitor displacements/drifts of the microscope components in close proximity to the sample and the MLL optics. In Fig. 1(c) it can be seen that, for example, for the sample stage (labeled 4 in the Figure) reflectors are mounted within 10 mm from the actual sample location.

The control system of the MLL microscope is based on the Experimental Physics and Industrial Control System (EPICS), which allows interdevice communication and control over the

network by way of the EPICS Channel Access protocol. A simple command-line interpreter named ECLI was developed on top of IPython, offering the full power of Python integrated with the EPICS devices in the system. Usage of the StepScan library allows for scans to be performed on all motors in the system, with the acquired data output to either HDF5 or a human readable (ASCII) format. Scan progress can then be visualized with a simple Matplotlib-based client, or alternatively the PyMca X-ray fluorescence toolkit. The Delta Tau Power PMAC, a Linux-based motion control system featuring the Xenomai real-time kernel, is the core of the motion control system. An extended PID (including feedforward terms) locks the X, Y, Z fine sample piezo scanner into position in

accordance with the AttoFPSensor interferometers digital quadrature encoder feedback. Combining the high-resolution digital-to-analog converter of the Power PMAC with the low-noise attocube ANC200 piezo amplifier, robust sub-nanometer positioning has been achieved. [10] All detectors in the system were triggered by the Struck SIS3820 VME scaler. During the initial synchrotron characterization experiments, fluorescence data were acquired via the Quantum Detectors Xspress 3 readout system and a Ketek fluorescence detector. Transmitted beam data for ptychographic reconstructions and Differential Phase Contrast data were acquired with the TimePix QTPX-262k detector from Amsterdam Scientific Instruments.

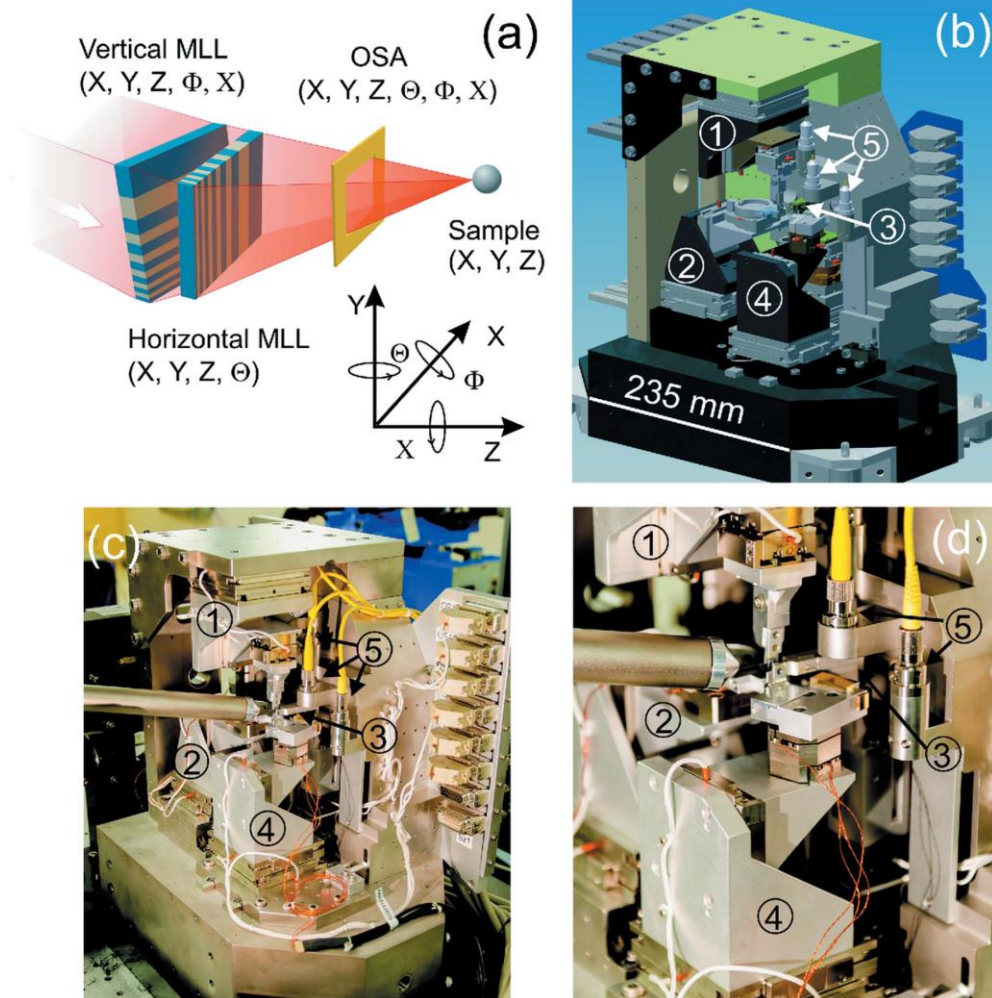


Figure 1. (a) Schematic of the MLL microscope. All required motions for individual components are shown. (b) CAD model of the MLL microscope. (c) Photograph of the assembled microscope, key components are enumerated: 1, vertical MLL assembly; 2, horizontal MLL assembly; 3, OSA assembly; 4, sample stage; 5, fiber optic interferometers heads. (d) Zoom-in of a sample area.

TEST OF THE MLL MICROSCOPE

The completed MLL microscope was installed and tested in the C-hutch of the Hard X-ray Nanoprobe (HXN) beamline at the NSLS-II, as shown in Fig 2. (a). The C-hutch is a remote satellite building designed for high vibrational and temperature stability (the decoupled foundation weighs about 500 tons and the temperature stability is better than 0.1 °C). Ptychography, fluorescence and DPC imaging experiments were performed under ambient conditions to characterize two dimensional nanofocusing of the MLL microscope. The beam defining slits selected coherent X-rays of 45 μm x 55 μm , which match the aperture of two

orthogonally aligned MLLs. Fluorescence images were acquired by raster scanning the sample and recording intensity of the L-edge line of Pt. The exposure time equals 200 ms with the pixel size of 5 nm. Figures 2b and 2c show the SEM and fluorescence images of the Pt test pattern respectively. Diameter of the 'donuts' is about 80 nm with the wall width of 20 nm. The foci were determined to be 12 nm and 13 nm for horizontal and vertical focal planes, respectively, which confirmed the high stability of the microscope. Overall drifts were on the order of 4 nm/hour based on the fluorescence image analysis.

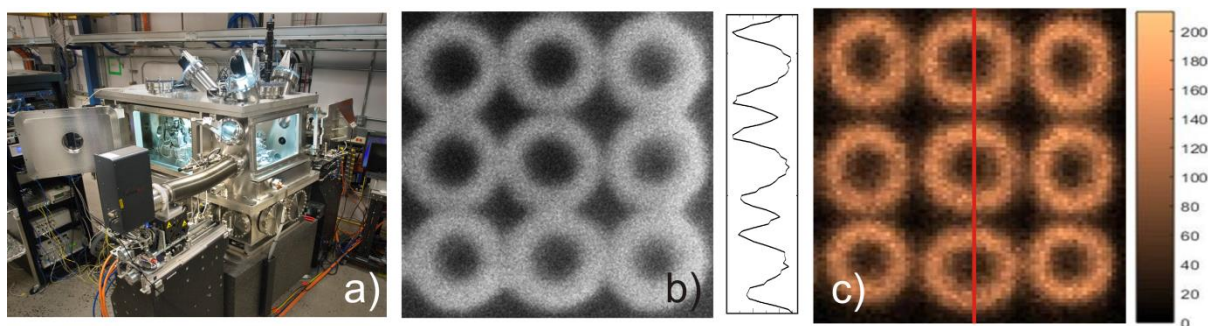


Figure 2. a): view of the MLL based microscope installed at the HXN beamline. b) SEM image of the Pt test pattern. Circles are 80 nm in diameter and 20 nm line width. The height of rings equals 200 nm. c) x-ray fluorescence image (platinum L-edge) of the same test pattern, with an x-ray energy of 12 keV, exposure time of 200 ms, and 5 nm per pixel. The scan profile along the horizontal line shown on the left demonstrates achieved spatial resolution.

BONDED MLL OPTICS FOR THE NEXT GENERATION MLL MICROSCOPE

In the previous section we introduced the first generation MLL X-ray microscope yielding 12 nm x 13 nm spatial resolutions x-ray imaging. In order to push the resolution even further more mechanical stability and less thermal drifts are required. This can be achieved by simplifying the microscope and reducing its footprint. This microscope we developed comprises of 4 major assemblies (horizontal MLL, vertical MLL, OSA and the sample stage). Both horizontal and vertical MLL assemblies function together to perform nm-scale accuracy alignment of two linear MLL optics. Overall, it results in five translational and three rotational degrees of motion with positions being maintained at the nm-scale level. The most complex angular motion is denoted as χ in Fig 1a and represents orthogonality between two lenses. [11] A misalignment of the former

introduces additional undesirable cross phase term in the x-ray wavefront which causes blurring effect on the focus. To achieve sub-10 nm point focus, the misalignment χ should remain below 0.01° for a typical MLL with a focal length of 4 mm at 12 keV photon energy and aperture size D of 40 μm . [12] Angular alignment must be maintained throughout the entire experiment. Alignment process can be simplified by bonding two linear MLLs together and fabricating of a monolithic 2D MLL optic. We have developed a dedicated setup and corresponding alignment procedure to perform proper bonding. The fact that χ angle is well defined allows installation of bonded MLL optics into a much simpler x-ray microscope with horizontal MLL and vertical MLL assemblies integrated, and makes it compatible with conventional ZP-based systems equipped with tip-tilt angular adjustment of the nanofocusing optics.

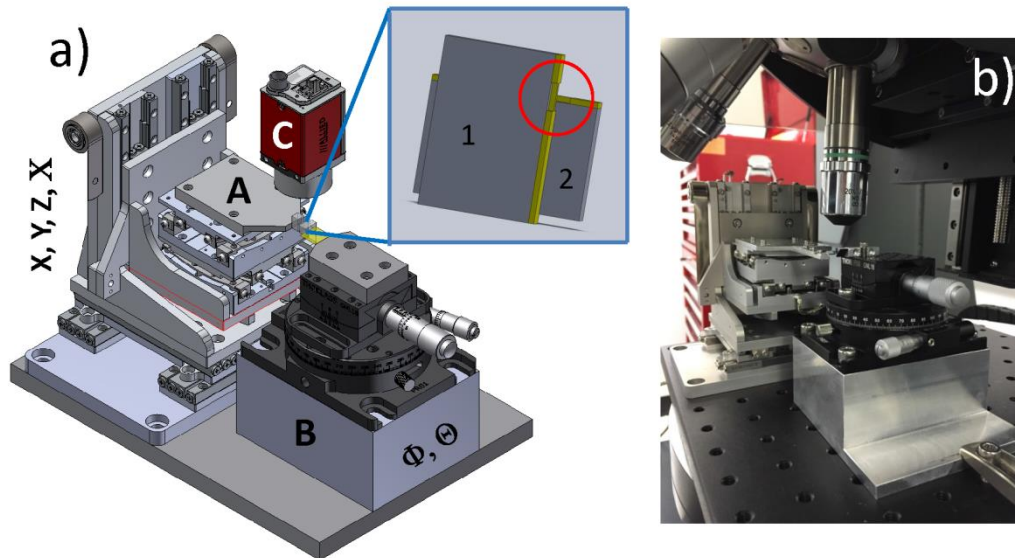


Figure 3. a) Computer Aided Design (CAD) model of the alignment system developed for MLL bonding. “A” represents manipulation system for vertical MLL and “B” for horizontal MLL, respectively. “C” schematically shows the head of a white light interferometer. The inset shows two bonded lenses with an overlapping MLLs region marked with a circle. Note, there is a redundant U motion in parts “A” and “B” used to optimize the view angle when aligning with respect to the interferometer head, b) photograph of the actual bonding setup.

The major challenge in preserving χ alignment during bonding process pertains to a small size of the lens itself. For a given length of MLL optic, 0.01° of angular tolerance χ translates into 175 nm height variation for every 1 mm of optic length. This is beyond resolution of any optical microscope, therefore we employed Zygo NewView 6000 3D optical surface profiler to characterize and correct misalignment errors during the bonding process (details of the bonding process in ref. [13]) Figure 4 demonstrates the optical microscope image of two bonded lenses (panel a), SEM image of the crossed MLLs (panel b) and surface profiles of two lenses measured by the white light

interferometer (panel c). Planes of the two lenses were measured to be separated by 221 μm which consists of thickness of the UV glue between two lenses and the recess of the FIB'd lens itself with respect to the Si substrate. The Ptychography measurements demonstrated linear focusing of the bonded MLL optics to be 12 nm and 24 nm in horizontal and vertical directions, respectively. [13] By optimizing the bonding process and utilizing higher resolution MLL optics, monolithic 2D nanofocusing devices should become available for the hard x-ray scanning microscopy community.

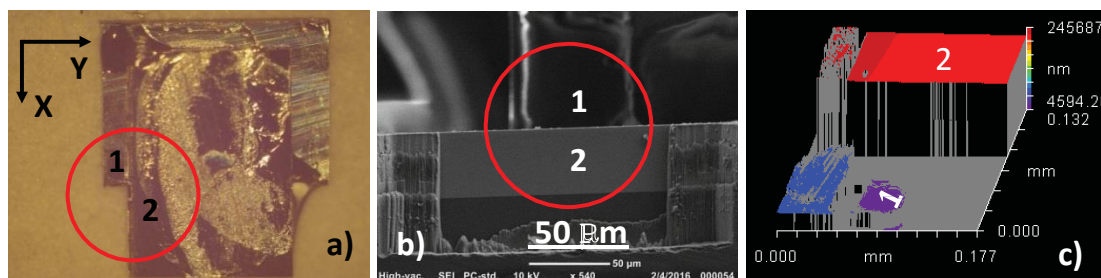


Figure 4. a) optical microscope image of two bonded MLL's. The overlapping area is marked with a circle. b) SEM image of two bonded lenses, c) optical surface profiler image of two bonded MLL's, the height difference between planes of two lenses equals 221 μm .

CONCLUSION

In this work, we present the first in its class scanning x-ray microscope based on the MLL nanofocusing optics. By reducing the form factor, reducing heat dissipation, developing high stiffness actuators, we successfully demonstrated fluorescence imaging with 12 nm x 13 nm spatial resolution at 12 keV. As a next step to push the resolution further, we have developed the prototype monolithic 2D MLL optics which is capable of producing point focus and significantly simplifies future generation MLL-based microscopy systems.

ACKNOWLEDGEMENT

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DESIGN OF A FLEXURE BASED XY PRECISION NANOPositionER WITH A TWO INCH TRAVEL RANGE FOR MICRO-SCALE SELECTIVE LASER SINTERING

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ABSTRACT

This paper presents the design of an XY nano-positioning flexure stage for the micro-scale selective laser sintering system (μ -SLS). The stage is designed to position and move the sample under the optical sub-system of μ -SLS machine and is designed to have a range of 2" with sub-micron accuracy. A modified double parallelogram flexure mechanism (DPFM) has been presented which eliminates the inherent under-constraint in DPFMs and alleviates the problems associated with it such as low in-plane uncontrollable resonance frequencies. Structural parameters have been designed to guarantee the desired range, stiffness and resonant frequency. The parametric design is verified by conducting finite element analysis (FEA), which reveals a reachable motion range over 2" in each working axis. Comparison between a typical DPFM stage and the modified DPFM stage on the basis of resonance frequencies has also been presented.

INTRODUCTION

The development of microscale additive manufacturing methods for metals and ceramics is important for a multitude of applications in the aerospace, electronics, and medical device industries. However, most commercially available metal additive manufacturing tools have feature-size resolutions of greater than 100 μm , which is too large to precisely control the microstructure of the parts they produce. One method to reduce the minimum feature-size resolution of metal additively-manufactured parts while still maintaining the throughput of traditional additive manufacturing processes is through the development a novel microscale selective laser sintering (μ -SLS) process [1,2]. In selective laser sintering, a layer of metal powders is spread into a powder bed and a laser is scanned across the powder bed to sinter together the metal powders in selected locations[3]. A new layer of powder is then spread on the bed and the process is repeated until a three dimensional structure is built up[2,4]. μ -SLS works in much the same way

except nanoscale powders, ultrafast lasers and scanning micro-mirror arrays are used to achieve microscale feature resolutions.

Figure 1 presents the schematic of the micro-scale selective laser sintering system. The XY nano-positioner in the system is an integral part of the μ -SLS system. The objective of this nano-positioner is to perfectly align the sample wafer under the objective lens of the optical system and to scan the sample under the micro-mirror based optical system with sub-micro accuracy and repeatability. Since, the micro-mirror based optical sub-system can only pattern an area of 2mm by 1mm at once, the wafer has to be moved under the sub-system to pattern the entire 2" by 2" area and hence the need to have the range of the system to be 2" along the X and Y directions. To realize a precision positioning with high accuracy, flexure-based compliant guiding mechanisms have been widely employed due to the absence of friction and backlash in these systems [5,6]. Selecting an appropriate actuator for this application is important to achieve the 2" range with sub-micron accuracy. Electromagnetic actuators or voice coil actuators (VCAs) are widely used in long-travel, flexure-based parallel XY stages since VCAs are capable of delivering a frictionless inch-level stroke along with a fine resolution.

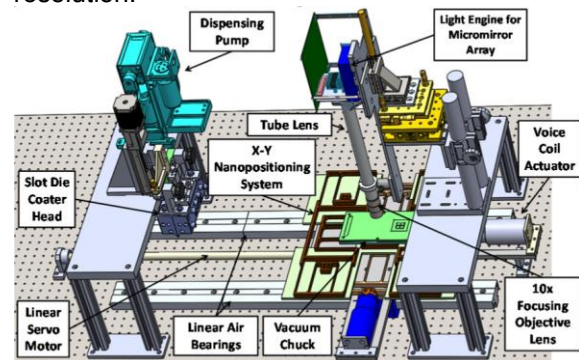


FIGURE 1. Schematic of the micro-SLS system

DESIGN OF XY FLEXURE STAGE

a) Design 1

Although the 2" stroke can be generated by selecting suitable actuators, it is challenging to devise a parallel flexure stage with both a large range and compact size simultaneously. To accomplish a large motion range, slender leaf flexures are usually employed[7]. The minimum width of the flexure is restricted by the practical tolerance of manufacturing and the maximum length is constrained by the compactness requirement[8].

A large number of two-dimensional flexure based parallel-kinematic positioning stages have been proposed by researchers. In previous works, double parallelogram flexure mechanisms (DPFMs) have been employed to achieve centimeter range motions [8].

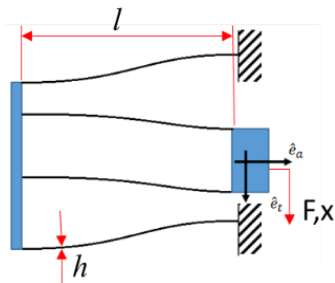


FIGURE 2. Schematic of a double parallelogram flexure mechanism (DPFM)

Applying the boundary conditions to the unit, the stiffness in tangential and axial directions is given by[9].

$$K_a \approx \frac{1}{\left(b^2 + \frac{9}{25}x_t^2\right)} \cdot \frac{12EI}{L} \quad (1)$$

$$K_t \approx \left[12 - \frac{3}{100} \left(\frac{F_a L^2}{EI}\right)^2\right] \cdot \frac{EI}{L^3} \quad (2)$$

where b , L , and E are the flexure width, beam length, and Young's modulus, respectively, l is the second moment of area of the flexure beam and F_a is the magnitude of axial force on the beam and x_t is the tangential displacement of the beam. For simplification, the axial force and tangential displacements can be assumed to be zero and the maximum displacement of the flexure beam is given by the following relation,

$$x_{max} = \sigma_y \cdot \frac{L^2}{3 \cdot E \cdot h \cdot FOS} \quad (3)$$

The maximum displacement of the stage is directly proportional to the square of length of the beams and inversely proportional to the thickness of the beams. The output motion in a particular direction is guided by four DPFMs located at the corners of the module. The second module is mounted orthogonally on the top of the first

module, resulting in a parallel-kinematic and stacked structure. This modular design allows for significant cost reduction in both manufacturing and maintenance. When driven by VCA1, four DPFMs on the top and two DPFMs at the bottom guide the motion along X and similarly for the motion along Y direction. Thus, a decoupled two-dimensional translation is achieved by the XY stage.

Using Al 7075 T6 as the material for the flexures, ($E = 71.7$ GPa, $\sigma_y = 503$ MPa, and density = 2810 kg/m³) with a factor of safety of 2 (to account for

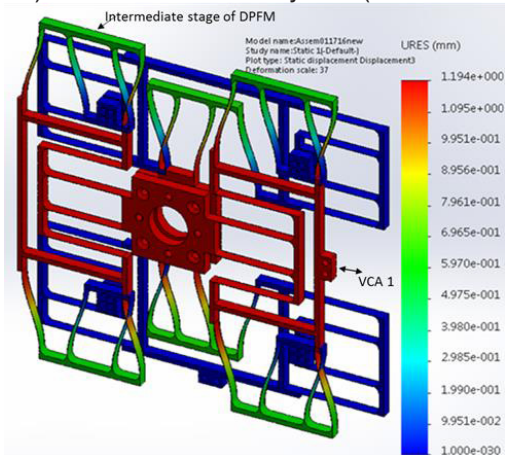


FIGURE 3. Static FEA simulation of deformation induced by an input force of 1N

TABLE 1. Structural parameters, stiffness and range of the stage using properties of Al-7075

Parameter	Value	Parameter	Value
l	101.4 mm	Kx	651 N/m
h	0.5 mm	Ky	651 N/m
b	12.7 mm	x_max	24.14 mm
FOS	2	Total range	48.28 mm

manufacturing tolerances, assembly stack up error and so on), x_{max} is calculated to be 48.28 mm. The FEA simulation conducted using SOLIDWORKS predicted a stiffness of 837.5 N/m in each working direction which is higher than the estimated stiffness of 651 N/m. This difference could be attributed to the addition of fillets (for ease of manufacturability). Using the values of stiffness and range from FEA simulation, the maximum actuation force needed to obtain the desired range was estimated to be 21.3 N. Based on the range and actuation force value, a suitable VCA with a maximum continuous force of 38.7N was selected for the stage. The maximum stress developed in the flexure stage was found to be 159.9 MPa which is 3 times lower than the yield strength of the material.

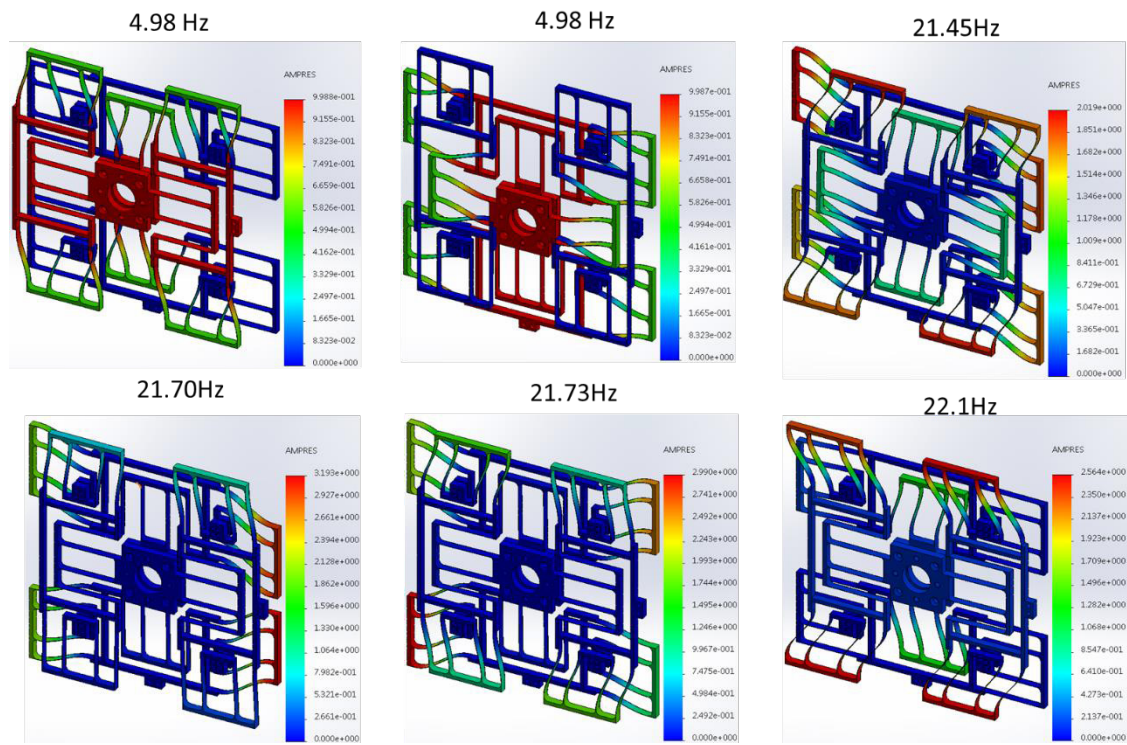


FIGURE 4. First-six resonant mode shapes and resonance frequencies of the XY stage

The maximum operating frequency of the system is 50Hz and thus, it is desired to have the uncontrollable resonant frequencies of the stage beyond 50 Hz. The first-six resonant frequencies and corresponding mode shapes as predicted by the modal analysis with SOLIDWORKS are illustrated in Figure 4. The FEA results reveal that the first-two mode shapes are contributed by the translations along the two working axes, respectively. The next four modes are the resonances of the individual DPFM units as shown in Figure 4. This suggests an uncontrolled DOF of the intermediate stage on the DPFM which causes the intermediate stage to be under-constrained as it is free to move even when the final stage is held fixed relative to the ground [9,10]. Although the first two modes are at 4.98 Hz, the center stage has VCAs attached to it along the two axes and hence, they can be controlled using appropriate feedback control action. However, individual DPFM units do not have any actuators attached to them which makes it difficult to control these modes of the DPFM. These modes can be damped passively by sandwiching a damping plate (a polymer plate with viscous fluid on either side or foam) between the two flexure planes as shown in Figure 5.

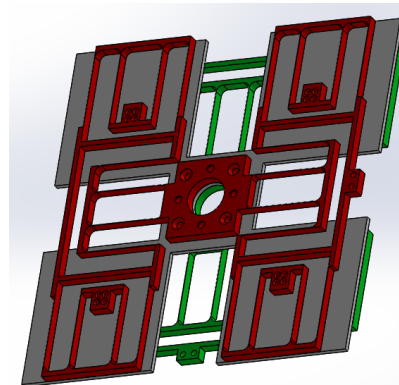


FIGURE 5. Damper plate in between the flexures

b) Design 2

A nested linkage design (see Figure 6) which eliminates the under-constraint in a DPFM unit has been presented by Panas and Hopkins [11]. The improved bearing design showed an 11 fold increase in the uncontrollable resonance frequency.

Figure 6 shows the design for the flexure stage with the nested linkage design for each of the DPFM units. The instant center of the added linkage is placed along the plane of symmetry of the structure.

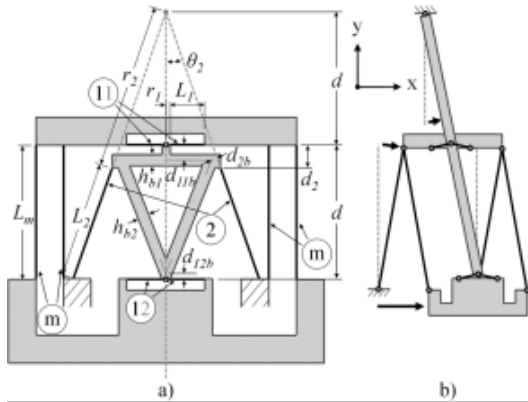


FIGURE 6. (a) An improved nested linkage design. The possible motion for the structure is shown in the equivalent linkage model in (b). The solid arrows show the nominal translational motion, which is equivalent to the DP flexure bearing alone [11]

The linkage guiding flexures (labeled 2 in Figure 6(a)) are nested within the central dead-space in the DPFM to reduce the footprint of the linkage. The linkage is attached to each of the stages with sets of blade flexures (labeled 11 and 12 in Fig. 6(a)) axially aligned to the final and intermediate stage DOF along the x-axis. These blade flexures are included symmetrically to avoid buckling, thermal sensitivity, and dynamic issues [11].

The improved nested linkage design maintains the same symmetry, and footprint as a regular DPFM unit via a small hollow linkage nested within the DPFM. However, the nested linkage mass is far from the center of motion which adds inertia roughly equivalent to the external linkage. The range, kinematics, and design of the main DOF are also effectively unaltered by the nested linkage design due to the design freedom possible for the linkage flexures. And thus, this improved linkage design can be employed in our application to improve the performance of our system. The thickness of the flexures has been reduced to 0.3mm from 0.5mm to achieve a similar stiffness as the previous design without increasing the overall footprint of the stage.

In this structure, the corner modules of both the top and bottom modules as shown in Figure 7 are mounted to the ground using a connector plate in between them. The middle flexures at the top module are connected to the center-stage of the bottom level and the middle flexures of the bottom module are connected to the top center-stage. The actuation method using VCAs remains the same.

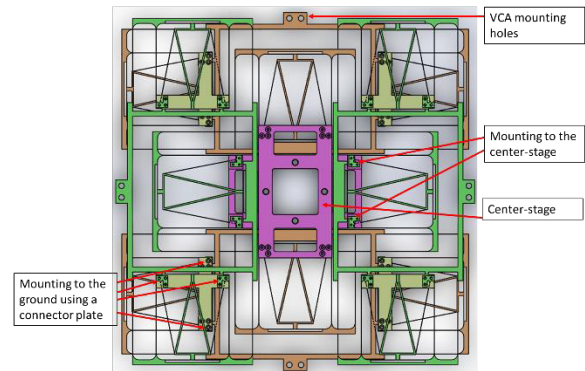


FIGURE 7. CAD model of Design 2 of XY stage

The FEA simulation predicted a stiffness of 853.2 N/m and thus, the maximum force requirement for the VCA becomes 21.5 N which is still lower than the maximum force achievable by the VCA selected above. The maximum stress developed in the flexure was found to be 337.7 MPa which equates to a factor of safety of 1.497 for the flexure stage.

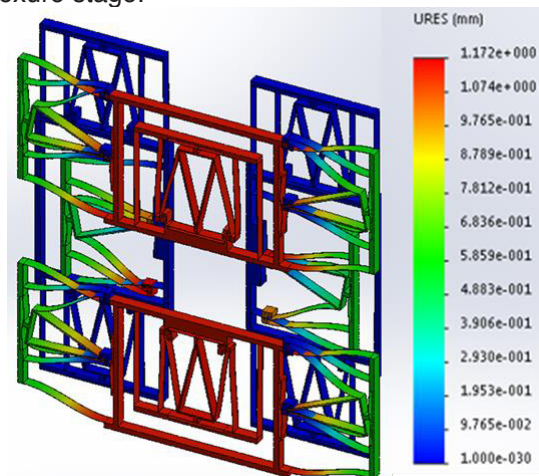


FIGURE 8. Static FEA simulation of deformation induced by an input force of 1N

In addition, the first-six resonant frequencies as predicted by the modal analysis are shown in Figure 9. The first-two mode shapes as expected are contributed by the translations along the two working axes, respectively. However, the higher modes of this stage are the out of plane resonances of the center-stage instead of the resonances of individual DPFMs which couldn't be controlled with the two VCAs. These out of plane resonances are beyond our operating range and hence, do not affect the performance of the stage during operation. It is interesting to note that the first resonance of individual DPFMs as observed with Design 1 doesn't appear even in the first 15 resonances. The first such resonance occurs at 169.5 Hz which is 8 times

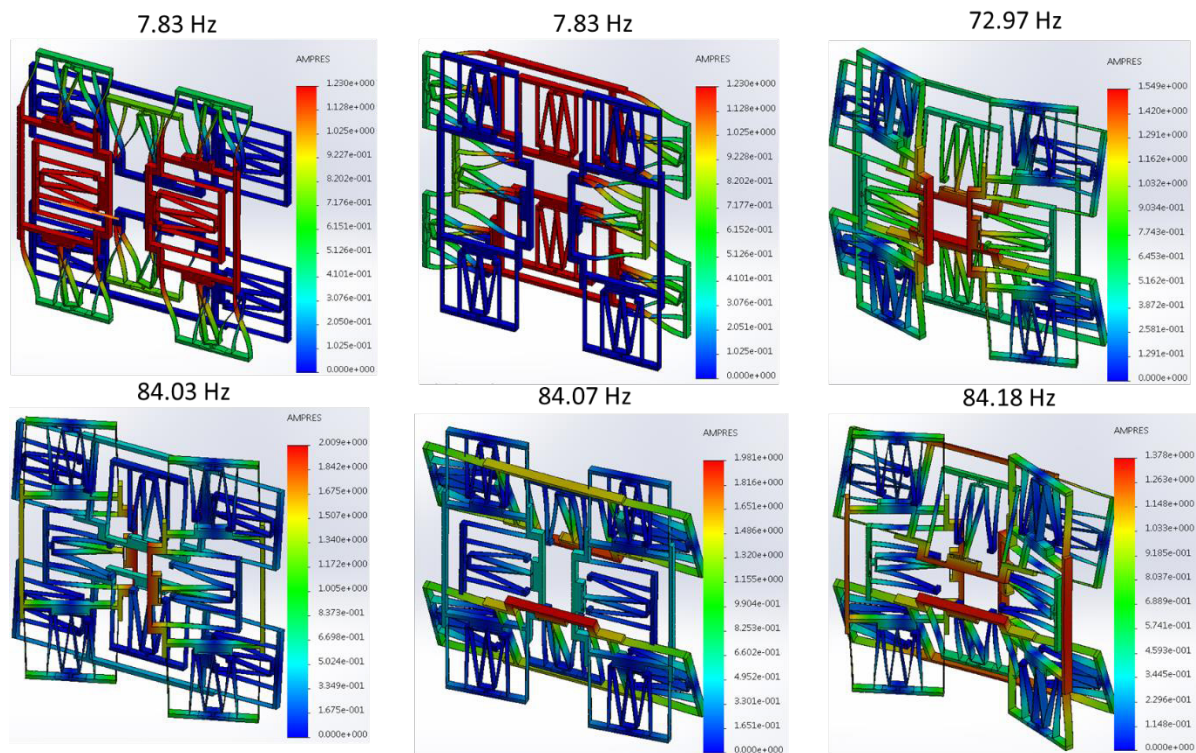


FIGURE 9. First-six resonant mode shapes and resonance frequencies of the design 2 of XY stage

the first in-plane resonance (21.45 Hz) of individual DPFMs observed with Design 1 showing that the under-constrained dynamics are observed to be raised by 8 times with the use of the nested linkage flexure design.

CONCLUSION

In this paper, the design of an XY nano-positioning stage with a range of 2 inches each working direction of the stage was presented. The paper discussed the parametric design which was substantiated by the static and dynamic FEA

results. A nested linkage design is shown through finite element analysis (FEA) to work as predicted in selectively eliminating the under-constrained degrees-of-freedom (DOF) in DP linear flexure bearings. The improved bearing shows an 8X gain in the resonance frequency of the under-constrained DOF. Further steps include assembly of the stage, developing a dynamic model of the stage, system identification of the stage to accurately understand the dynamic behavior and finally, developing a robust control scheme for the system using the model obtained from system identification.

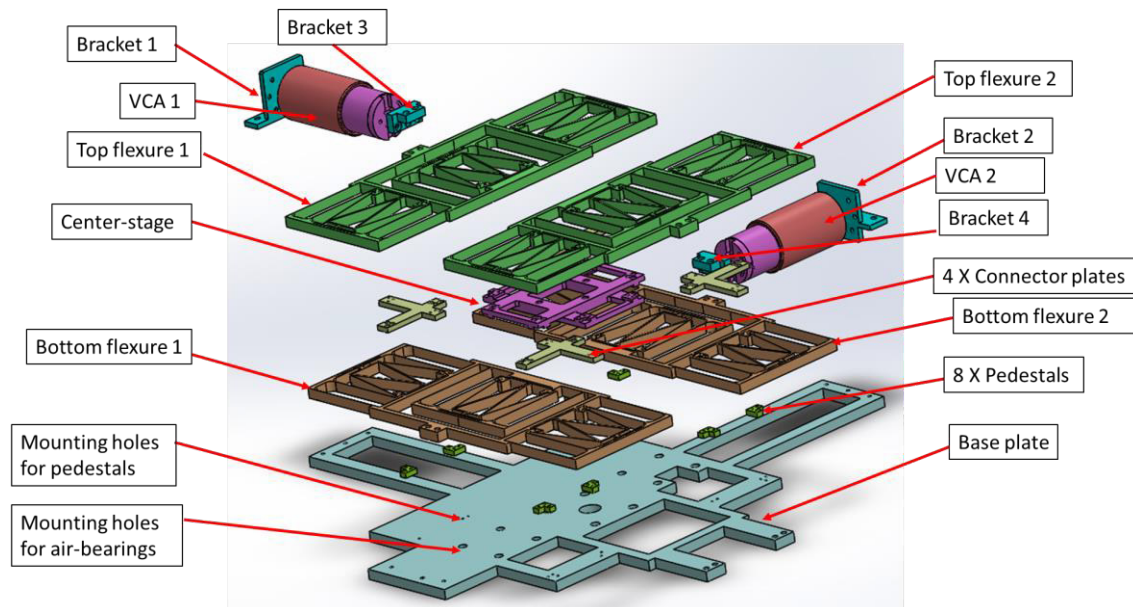


FIGURE 10. Exploded view of CAD model of XY stage assembly

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Micro-Mirror Array: Classification, Review, and Performance Comparison

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ABSTRACT

The aim of this paper is to provide a comprehensive overview of MMA technologies and their fundamental characteristics, and to compare and analyze the performance of MMA designs. General classifications of MMA devices are introduced. Performance metrics are defined and their practical limits are estimated based on actuation physics. Scatter plots are provided to compare the performance of all existing two-dimensional (2D) MMA devices that achieve at least tip and tilt motion.

MMA DEVICE CLASSIFICATION

Micro-mirror array are microelectromechanical systems constructed with micro-size movable mirrors arranged in planar array. The mirrors are orientated largely parallel to the planar array, where each can be driven to realize rotational or translational motion within certain range. In general, MMAs can be classified based on the following characteristics.

1) *Array dimension*

Most MMA devices are two-dimensional (2D) arrays, i.e., the mirrors are arranged to cover a planar area of similar length and width. Some MMA devices are one-dimensional (1D) arrays that forms a long narrow band. In a general sense, single micro-mirror devices can be classified as zero-dimensional (0D) arrays.

2) *Mirror surface type*

The mirror surface of an MMA can be either discrete or continuous. In a discrete-surface MMA, the surface of each single micro-mirror are separated, and the motion of a single mirror has minimal influence on the adjacent mirrors. In a continuous-surface MMA, all micro-mirror units share a whole mirror surface (usually a membrane), resulting in a 100% fill factor. The mirror units are not able to do individual cell rotation as with discrete mirrors. Rather they drive localized piston deflections in the membrane.

3) *Degree of freedom (DOF) of mirror motion*

There are three DOFs of single mirror motion that are optically significant: tip, tilt and piston. Accordingly, MMAs are classified into five types: Tip Only (T), Piston Only (P), Tip and Tilt (TT), Tip and Piston (TP), Tip Tilt and Piston (TTP).

4) *Actuation form*

The actuation of micro-mirror motion can be realized in three forms of signal: analog[1,2], digital[3,4] and resonant[5,6]. An analog-actuation micro-mirror is actuated by a continuously varying signal, which has a 1:1 mapping to mirror position. This allows the mirror to statically reach any position within the motion range. A digital-actuation micro-mirror is actuated by discrete and limited set of voltages. It operates at only a few stable positions (usually two). A resonant-actuation micro-mirror is actuated by a repetitively varying signal that often has the same frequency as the device's mechanical resonance frequency. It is understood to be a fundamentally dynamic system, so the range is quoted at the resonant operation.

5) *Actuation Physics*

There are five basic actuation physics types for MMA: electrostatic plate, electrostatic comb, thermal actuation, Lorentz actuation and piezoelectric actuation. However, the actuator design and performance depend largely on mirror surface type, thus for this research MMA actuation physics is divided into eight categories: Plate Discrete, Comb Discrete, Thermal Discrete, Lorentz Discrete, Piezo Discrete, Plate Continuous, Piezo Continuous, and Lorentz Continuous (There is no existing Comb Continuous or Thermal Continuous MMA designs).

MMA PERFORMANCE METRICS

In order to compare the performance of micro-mirror devices quantitatively, the following metrics are defined universally for all micro-mirror

¹These are both first authors as their contributions to this paper are equal

designs:

1. Characteristic length L is defined to describe the size of unit micro-mirror. For micro-mirrors in array, it is defined as the pitch, i.e., the average spacing between two adjacent mirrors. For single micro-mirrors, it is defined as the square root of mirror surface area.

2. Motion range θ_{total} and d_{total} are defined as the total range of mirror tip/tilt/piston motion that can be achieved statically by analog actuation. The half motion range θ_{act} and d_{act} , or sometimes referred to as actuation range in later sections, are defined to be the maximum rotation angle or displacement from equilibrium position.

3. Maximum acceleration α_{max} and a_{max} are defined as the maximum angular/linear acceleration of mirror motion that is achievable during actuation. Under second-order system assumption, the maximum angular acceleration α_{max} is calculated as follows:

$$\alpha_{max} = \frac{\tau_{max}}{J} = \frac{\gamma_{\theta} \tau_{max\theta}}{J} = \frac{\gamma_{\theta} \cdot k_{\theta} \theta_{act}}{J} \quad (1)$$

$$= \gamma_{\theta} \theta_{act} \omega_{\theta n}^2$$

where τ_{max} is the maximum torque output of the micro-actuator, J is the moment of inertia of the mirror, $\tau_{max\theta}$ is the torque needed for static maximum tilting, γ_{θ} is the ratio between τ_{max} and $\tau_{max\theta}$, k_{θ} is the rotational spring constant, θ_{act} is the maximum tilting angle from equilibrium position, and $\omega_{\theta n}$ is the resonant frequency of tilting motion. Maximum piston acceleration a_{max} can be calculated similarly.

4. Mirror energy E_{max} is defined as the maximum kinetic energy of mirror under sinusoidal excitation within the motion range. This metric describes the amount of energy that can be utilized by one micro-mirror unit. For tipping/tilting motion, it is calculated as follows:

$$E_{\theta max} = \frac{1}{2} J (\dot{\theta})_{max}^2 \quad (3)$$

where $(\dot{\theta})_{max}$ is the maximum angular speed of mirror motion, and can be derived using a simple

derivative of a cosine trajectory as:

$$(\dot{\theta})_{max} = \theta_{act} \omega_{\theta max} = \theta_{act} \sqrt{\frac{\alpha_{max}}{\theta_{act}}} \quad (4)$$

$$= \sqrt{\alpha_{max} \theta_{act}}$$

where $\omega_{\theta max}$ is the angular frequency when angular speed achieves the maximum value.

A rough approximation of the mirror kinetic energy scale for tipping/tilting motion becomes:

$$E_{\theta max} = \frac{1}{2} J (\dot{\theta})_{max}^2 \quad (6)$$

$$= \frac{1}{2} C_1 C_2 \rho_m (\gamma_{FF} \cdot L)^5 \alpha_{max} \theta_{act}$$

where C_1 is the moment of inertia coefficient that depends on the geometry of the mirror, C_2 is the ratio between mirror thickness and side length, γ_{FF} is square root of MMA fill-factor ($\gamma_{FF} = 1$ for single micro-mirror device), L is the characteristic length of the mirror array, and ρ_m is the density of mirror material. Mirror kinetic energy scale for piston motion can be calculated similarly.

When comparing the mirror performance, the approximate values of $C_1 = 1/12$, drawn from the rotation of square plates, and $C_2 = 10^{-2}$ drawn from aspect ratios on existing mirror designs, are used for all designs.

5. The mirror energy density is defined as

$$U_{max} = \frac{E_{max}}{L^3} \quad (8)$$

This metric demonstrate the effectiveness of mirror actuation by eliminating the factor of mirror size.

MMA ACTUATION PERFORMANCE LIMIT

In this section, the equations of energy density limits for tipping and tilting motion are derived based on the utilizable energy of each actuation physics. Numerical values of energy density limit are estimated for MMAs with both rotational DOFs. These limits are plotted and compared against the performance of all 2D TT and TTP MMAs in the next section. The energy density

limit for piston can be estimated using similar process, but it will not be discussed in this paper.

1) Plate Actuation Limit

The usable actuation energy of electrostatic plates can be derived as:

$$\begin{aligned} E_{plate} &= \tau_{plate} \cdot \theta_{act} \\ &= \frac{\varepsilon V_{max}^2}{2d_{gap}^2} \cdot L^3 \cdot \gamma_L \cdot \gamma_A \cdot \theta_{act} \end{aligned} \quad (9)$$

where ε is the electric constant; V_{max} is the maximum voltage applied on electrostatic plates; d_{gap} is the separation between electrodes; γ_L is the lever factor, defined as the ratio between the length of the actuation lever arm and characteristic length; and γ_A is the actuation area factor, defined as the ratio between the usable area for actuation and mirror area.

The energy density of plate actuation:

$$U_{plate} = \frac{E_{plate}}{L^3} = \frac{\varepsilon V_{max}^2}{2d_{gap}^2} \cdot \gamma_L \cdot \gamma_A \cdot \theta_{act} \quad (10)$$

d_{gap} is estimated based on geometric relationship between plate separation distance and mirror tilting range. Large actuation range requires large separation between two electrostatic plates. In a typical plate actuation MMA design, d_{gap} are related to θ_{act} as follows:

$$d_{gap} = \theta_{act} \gamma_P \gamma_L L \quad (11)$$

where γ_P is the pull-in factor, defined as the ratio between the plate separation and maximum plate displacement without pull-in. Typically γ_P is 3 or larger.

The practical limit of energy density for Plate MMAs with both tip and tilt DOFs is estimated to be:

$$U_{plate} = 4.43 \times 10^{-8} \cdot \frac{1}{\theta_{act} L^2} \quad (12)$$

It can be seen that U_{plate} decreases when θ_{act} and L increase, suggesting that, for Plate MMA design, achieving better dynamic performance is

in conflict with achieving large actuation range and large mirror size.

2) Comb Actuation Limit

The usable actuation energy of electrostatic combs can be derived as:

$$\begin{aligned} E_{comb} &= f_{comb} \cdot A_{comb} \cdot L_{level} \cdot \theta_{act} \\ &= \frac{\varepsilon V_{max}^2}{d_{gap}} \cdot \frac{\gamma_A L^2}{2d_{gap} \gamma_d} \cdot \gamma_L \cdot \theta_{act} \end{aligned} \quad (13)$$

where γ_d is the comb finger gap factor, defined as $\frac{d_{pitch}}{2d_{gap}}$, the inverse of the fraction of the pitch

used by the gap. The first term describes the electrostatic shear motion force density, the second term describes the top down area of combs divided into repeating sets by the pitch, the third term describes the effective lever arm and the fourth term covers the range of motion.

The energy density of comb actuation:

$$U_{comb} = \frac{E_{comb}}{L^3} = \frac{\varepsilon V_{max}^2}{2d_{gap}^2 \gamma_d} \cdot \gamma_L \cdot \gamma_A \cdot \theta_{act} \quad (14)$$

In contrast to plate actuation, comb actuation allow the comb finger gap to be largely independent of the actuation range. Typically, the smallest value d_{gap} can achieve is on the order of $2\mu\text{m}$.

The practical limit of energy density for Comb MMAs with both tip and tilt DOFs is estimated to be:

$$U_{comb} = 3113 \theta_{act} \quad (15)$$

The above equation suggests that, for Comb MMA design, the energy density limit is higher when actuation range increases, but not affected by mirror scaling.

3) Piezo Actuation Limit

The usable actuation energy of piezoelectric actuators can be derived as:

$$E_{piezo} = \frac{1}{2} V_{piezo} E_{piezo} \varepsilon_{max}^2 \quad (16)$$

where V_{piezo} is the volume of piezoelectric material, E_{piezo} is the Young's modulus of piezoelectric material, and ε_{max} is the maximum achievable strain during actuation.

Lead Zirconate Titanate (PZT) is most widely used for piezoelectric actuation. The maximum allowable electric field on PZT is under 2kV/mm, which generates up to 0.2% strain[7,8]. We define the volume factor γ_V as V_{act}/L^3 , whose value typically varies between 1×10^{-5} and 5×10^{-4} for Piezo MMAs [9–13].

The practical limit of energy density for Piezo MMAs with both tip and tilt DOFs can be estimated as:

$$U_{piezo} = \frac{E_{piezo}}{L^3} = \frac{1}{2} \gamma_V E_{PZT} \varepsilon_{max}^2 \quad (17)$$

$$= 70 \text{ J} \cdot \text{m}^{-3}$$

The above equation is simply an estimate of Piezo MMA energy density limit. The value of γ_V depends on the design of mirror structure.

4) Thermal Actuation Limit

The usable actuation energy of thermal actuation can be derived as:

$$E_{thermal} = \frac{1}{2} V_{thermal} E_{thermal} \varepsilon_{max}^2 \quad (18)$$

where $V_{thermal}$ is the volume of thermal actuation material (typically the material with larger thermal expansion coefficient in the unimorph or bimorph) and $E_{thermal}$ is the Young's modulus of thermal actuation material.

Aluminum is usually chosen as thermal actuation material for its large thermal expansion coefficient comparing to silicon. During actuation, the temperature increase of thermal actuator is 60K~300K, which generates up to 0.6% strain. The value of γ_V for Thermal MMAs typically varies between 1×10^{-5} and 1×10^{-4} [14–19].

The practical limit of energy density for Thermal MMAs with both tip and tilt DOFs can be estimated as:

$$U_{thermal} = \frac{E_{thermal}}{L^3} = \frac{1}{2} \gamma_V E_{Al} \varepsilon_{max}^2 \quad (19)$$

$$= 124 \text{ J} \cdot \text{m}^{-3}$$

Similar to Piezo MMA, this value is merely an estimate of Thermal MMA energy density limit.

5) Lorentz Actuation Limit

The usable actuation energy of Lorentz actuation can be derived as:

$$E_{Lorentz} = \tau_{Lorentz} \cdot \theta_{act}$$

$$= \frac{N \gamma_A \gamma_L B I}{d_{pitch}} L^3 \cdot \theta_{act} \quad (20)$$

where N is the number of current trace layers, B is the magnetic field strength, I is the electric current, and d_{pitch} is the pitch between wire traces. The actuation energy density:

$$U_{Lorentz} = \frac{E_{Lorentz}}{L^3} = \frac{N \gamma_A \gamma_L B I}{d_{pitch}} \theta_{act} \quad (21)$$

For Lorentz MMA devices, Joule heating and heat dissipation are the main constraints limiting the current. To avoid overheating, joule heating rate should not exceeds the maximum allowable heat dissipation rate when current reaches the maximum steady value. And the following inequality can be derived:

$$\frac{I}{d_{pitch}} \leq \sqrt{\frac{h \gamma_s \Delta T_{max} t_{trace}}{\gamma_d \gamma_A \rho_\Omega}} \quad (22)$$

where h is the heat transfer coefficient of mirror surface; γ_s is surface area factor, defined as

$\frac{A_{Surface}}{A_m}$, the ratio between the total surface area

and mirror area; ΔT_{max} is the maximum temperature difference between the device temperature and ambient temperature; t_{trace} is layer thickness of trace; γ_d is the trace gap factor,

defined as $\frac{d_{pitch}}{d_{gap}}$, the inverse of the fraction of

the pitch used by the gap; and ρ_Ω is resistivity of trace material.

The energy density limit of Lorentz actuation:

$$U_{Lorentz} = \frac{N\gamma_A\gamma_L BI}{d_{pitch}} \theta_{act} \quad (23)$$

$$= N\gamma_L B \sqrt{\frac{h\gamma_s\gamma_A \Delta T_{max} t_{trace}}{\gamma_d \rho_\Omega}} \cdot \theta_{act}$$

The practical limit of energy density for Lorentz MMAs with both tip and tilt DOFs can be estimated to be:

$$U_{Lorentz} = N\gamma_L B \sqrt{\frac{h\gamma_s\gamma_A \Delta T_{max} t_{trace}}{\gamma_d \rho_\Omega}} \cdot \theta_{act} \quad (24)$$

$$= 945\theta_{act}$$

For Lorentz MMA design, the energy density limit scales with the actuation range.

MMA PERFORMANCE PLOTS

In this section, the performance of 87 2D TT and TTP MMA designs are plotted and compared. Figure 2 shows the legend used in all scatter plots to represent the mirror types of every scatter dots.

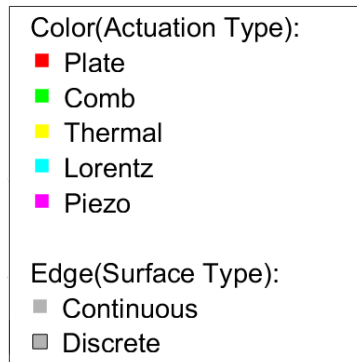


Figure 2. Legend of Scatter Plots

Figure 3 compares the tilting motion range, characteristic length and maximum angular acceleration. Figure 4 compares the energy density of MMAs as well as the estimated limits at $\theta_{act} = 3^\circ$ for all actuation types. Figure 5 compares the mirror surface size, fill factor and characteristic length.

CONCLUSIONS

The existing micro-mirror technologies have been categorized and analyzed. The performance evaluation metrics of MMA designs have been introduced. Plots that compare 2D TT and TTP MMA performance have been generated and rough bounds on theoretical limits have been provided to help designers choose the optimal

actuation method and recognize feasible performance capabilities.

This work was performed in part under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344. LLNL-ABS-686644.

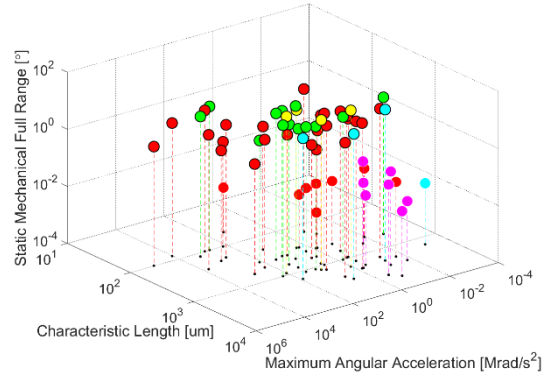


Figure 3. Range vs. Pitch vs. Acceleration

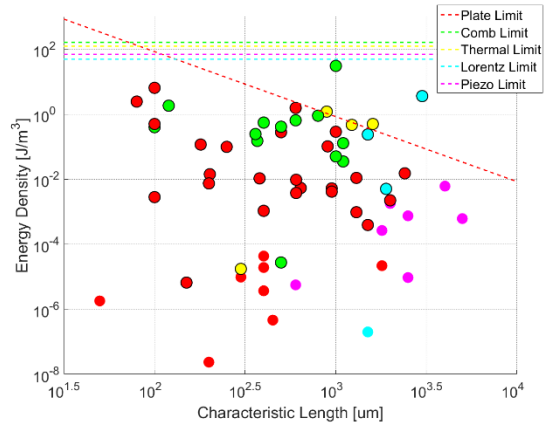


Figure 4. Energy Density vs. Pitch

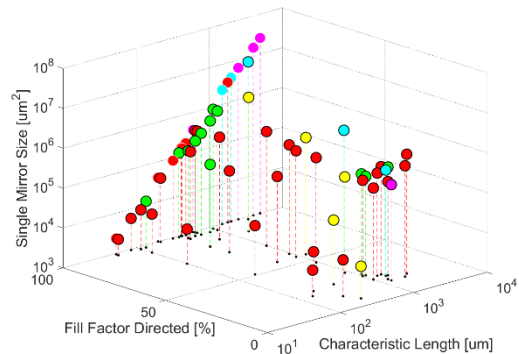


Figure 5. Mirror Size vs. Fill Factor vs. Pitch

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Omnidirectional One-Shot Exposure Micro-Stereolithography for Biomimetic Hemisphere with Micro Structured Surface

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INTRODUCTION

Certain creatures' functional micro structures are known as useful in human life and in industry. Especially, creatures' special surfaces with complex and fine structures are attracting attention because they are available in various industrial situation. For example, mosquito's eyes have water repelling and anti-reflection functions on their surface (*FIGURE 1*) [1][2], and leaves of lotus are known as their water repelling and self-cleansing structures [3]. Generally, most of creatures' functional surfaces consist of a lot of several tens of micrometers hemispheres and a lot of micro structures on each hemisphere. They are highly complex and fine structures. Conventional methods such as photolithography, nanoimprint, micro-stereolithography and focused ion beam have difficulty to make such structures without taking much time or spending high cost. More improved fabrication techniques are needed to apply making biomimetic hemisphere with micro structured surface.

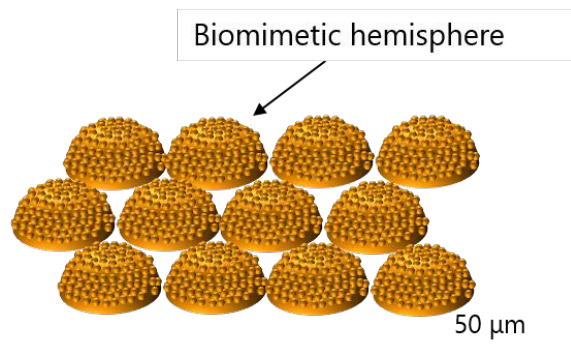


FIGURE 1. Scheme of example of a biomimetic hemisphere of which surface micro structures are arrayed.

In this research, we proposed omnidirectional one-shot exposure micro-stereolithography to fabricate creatures' complex and fine structures. Radial light from spot light source on substrate surface is propagated as a spherical wave. When

surrounding area is filled with photosensitive resin, the light is able to solidify resin with hemisphere shape. If the angle intensity distribution of the light is uneven, the surface of solidified resin shape is also uneven. In this method controlling angle distribution can make the biomimetic hemisphere which has fine uneven structures on its surface. It is suitable for fabricating needed structures. This report explains the concept of omnidirectional one-shot exposure micro-stereolithography, numerical simulation of this method and the examination by basic experiments.

CONCEPT OF OMNIDIRECTIONAL ONE-SHOT EXPOSURE

Fabricating Hemisphere

When light passes through a boundary between two different mediums from high refractive index medium to low one, a refracting angle is larger than an incident angle. Therefore light focused on the boundary with high numerical aperture is transmitted from focused spot to every direction into low refractive index medium. When photosensitive resin is filled as a low refractive index medium, the omnidirectional transmitted light is expected to solidify the resin and make it a hemisphere structure (*FIGURE 2*).

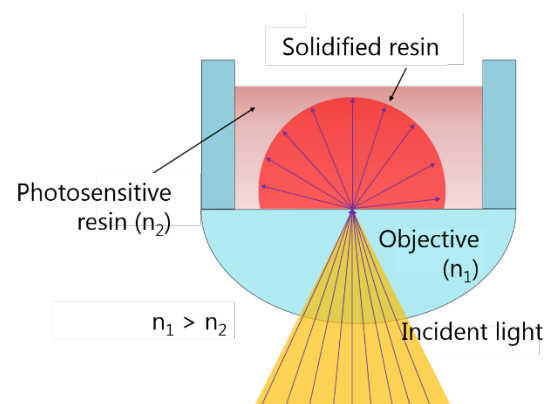


FIGURE 2. Proposed omnidirectional one-shot exposure micro-stereolithography.

Fabricating Surface Micro Structures

The light intensity distribution changed according to each incident angle can cure resin with fine uneven structures on the hemisphere surface (FIGURE 3). This method allows us to fabricate biomimetic hemisphere with micro structured surface in a one-shot process.

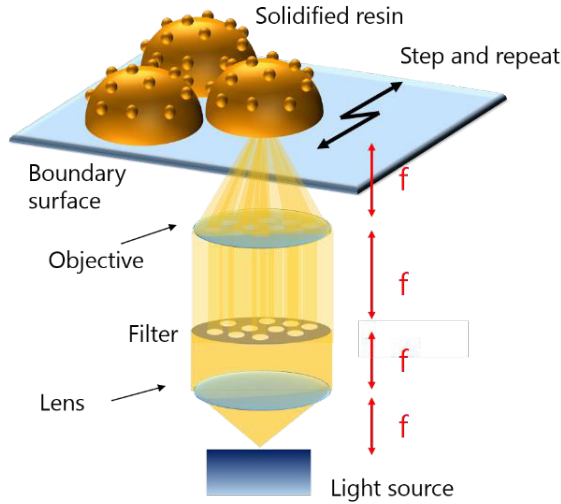


FIGURE 3. Fabricating biomimetic hemisphere with micro structured surface by controlling incident angle distribution.

NUMERICAL SIMULATION

In this section, method and results of numerical simulation are explained. Developed simulator allows us to get theoretical solidified shape. Some simulations were conducted to confirm principles and utility of this method.

Simulation Method

In this method, incident angle distribution is changed by controlling spacial intensity distribution in plane wave before an objective as shown in FIGURE 3. The variation of the area density of energy should be considered before and behind the objective. The light turns from a plane wave to spherical surface wave at the objective. Also, the energy density is changed at the point of the boundary between two mediums. The area of spherical surface is expanded by Snell's law because refractive index of resin is more than that of the objective side. FIGURE 4 shows the changes of areas of wave surfaces. In this FIGURE 4, r is the radius of the plane wave, θ_i is incident angle of the light of its small area, θ_t is transmission angle of the light of its small area, n_1 and n_2 are refractive indexes of the incident side medium and transmission side medium

respectively ($n_1 > n_2$), f is focal length of the objective, R is distance from the focal point of objective which is on the boundary surface. In conclusion, total area ratio is as follows.

$$\begin{aligned} & \frac{n_2^2 R^2 \sin \theta_t d\varphi d\theta_t}{r dr d\varphi} \\ &= \frac{n_2^2 R^2 \sin \theta_t d\theta_t}{r d\theta_i} \frac{d\theta_i}{n_1 f (\sin(\theta_i + d\theta_i) - \sin \theta_i)} \\ &= \frac{n_2^2 R^2 \sin \theta_t}{n_1 f \sin \theta_i} \frac{1}{n_1 f \cos \theta_i} \frac{d\theta_t}{d\theta_i} \\ &= \frac{n_2^2 R^2}{n_1^2 f^2 \cos \theta_i} \frac{n_1 \cos \theta_i \sin \theta_t}{n_2 \cos \theta_t \sin \theta_i} \\ &= \frac{R^2}{f^2 \cot \theta_t} \end{aligned} \quad (1)$$

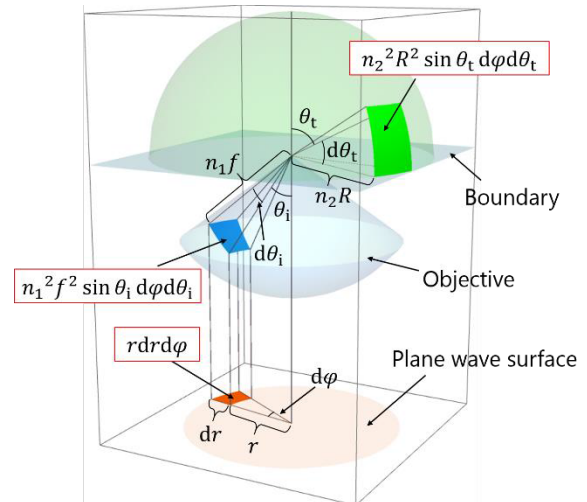


FIGURE 4. Variation of area density of intensity at the respective boundaries.

The intensity of the light in the transmission side is influenced by transmission coefficients derived by Fresnel's formula [4].

$$\begin{aligned} t_s &= \frac{2n_1 \cos \theta_i}{n_1 \cos \theta_i + n_2 \cos \theta_t} \\ t_p &= \frac{2n_1 \cos \theta_i}{n_2 \cos \theta_i + n_1 \cos \theta_t} \end{aligned} \quad (2)$$

where, t_s is the coefficient of s-polarization, t_p is the coefficient of p-polarization. Amplitude of wave from the boundary is multiplied by these coefficients depending on its polarization. In this case, these coefficients are squared to be multiplied by intensity, and averaged because

polarization is random. The light intensity in resin is attenuated by light absorption of resin. It is described by Lambert-Beer law [5].

$$I(x) = Ie^{-\alpha x} \quad (3)$$

where, I represents light intensity on the boundary surface, $I(x)$ represents light intensity at x distant from the point of transmission, α is absorption coefficient depending on resin. In conclusion, Intensity in resin is described as follows.

$$I(R) = I_0 \frac{f^2 \cot \theta_t (t_s^2 + t_p^2)}{R^2} e^{-\alpha R} \quad (4)$$

where, I_0 is intensity of incident plane wave.

Simulation Result

As conditions, a refractive index of incident side is 1.78, a refractive index of transmission side is 1.5, a numerical aperture of the objective is 1.65 and its focal length is 1.8 mm. Absorption coefficient α is 1000 which is a general value.

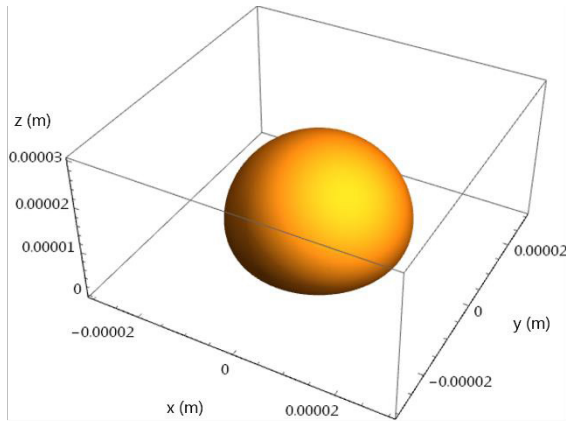


FIGURE 5. Solidified resin shape made by omnidirectional one-shot exposure.

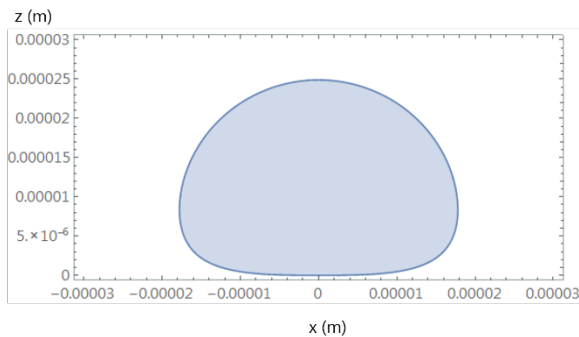


FIGURE 6. Sectional view of solidified shape.

First, we confirmed the shape of solidified resin made by this method without any filter. FIGURE 5 shows 3 dimensional plot of solidified resin shape made by omnidirectional one-shot exposure. FIGURE 6 is a sectional view of it. Although there is a little space at edge of the structure, entire shape resembles a hemisphere structure. It was verified that a structure like a hemisphere is able to be cured by this method. Second, we simulated making micro structured surface on the hemisphere. A filter with a lot of holes as shown in FIGURE 7 was used. 100 % intensity is transmitted at points of white small holes, and at grey place 80 % intensity is transmitted. FIGURE 8 is 3 dimensional plot, and FIGURE 9 is sectional view. These results verified that a biomimetic hemisphere is able to be fabricated by this method.

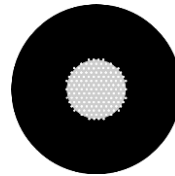


FIGURE 7. Filter with a lot of holes.

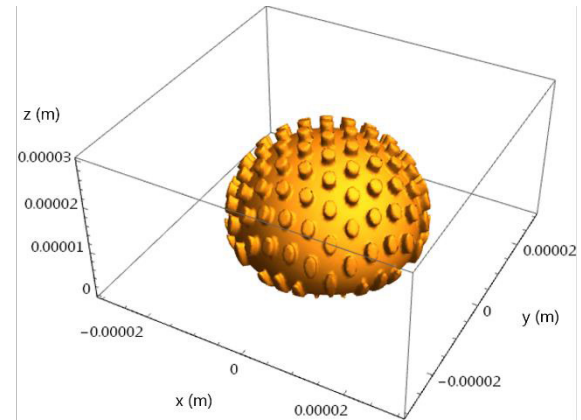


FIGURE 8. Solidified resin shape made by one-shot exposure with filter.

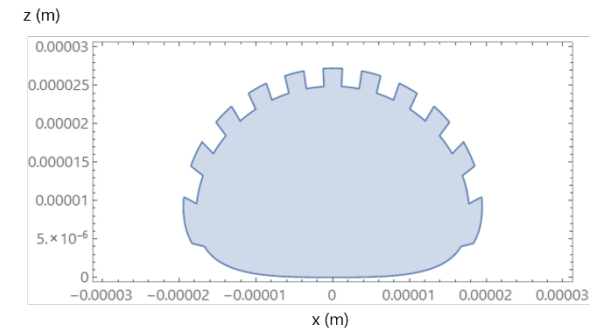


FIGURE 9. Sectional view of the shape with filter.

DEVELOPMENT OF APPARATUS

FIGURE 10 shows omnidirectional one-shot exposure system. The used objective has high numerical aperture (NA=1.65) and high refractive index medium ($n=1.78$). It was used as a solid immersion lens by combining with immersion oil ($n=1.78$) and cover glass ($n=1.78$). The used photosensitive resin's refractive index is 1.5. A filter was used to change incident light angle distribution. The filter should be set at the objective's back focal plane. However, it is impossible because that place is inside of the objective. An optical system was developed for the filter image to occur at the inside back focal plane. Three lenses and objective consist to two pairs of infinity correction optical system. The filter was set between lens 1 and lens 2.

RESULTS OF EXPERIMENT

As a light source, LED light with 470 nm wavelength and aperture were used. FIGURE 11 shows SEM images. Light energy was controlled by exposure time. Intended complex and fine structure was able to be fabricated.

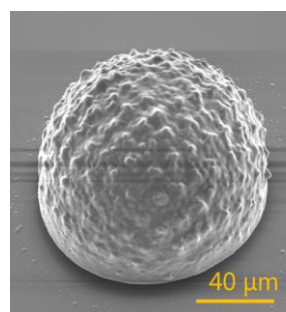


FIGURE 11. SEM images of omnidirectional one-shot exposure

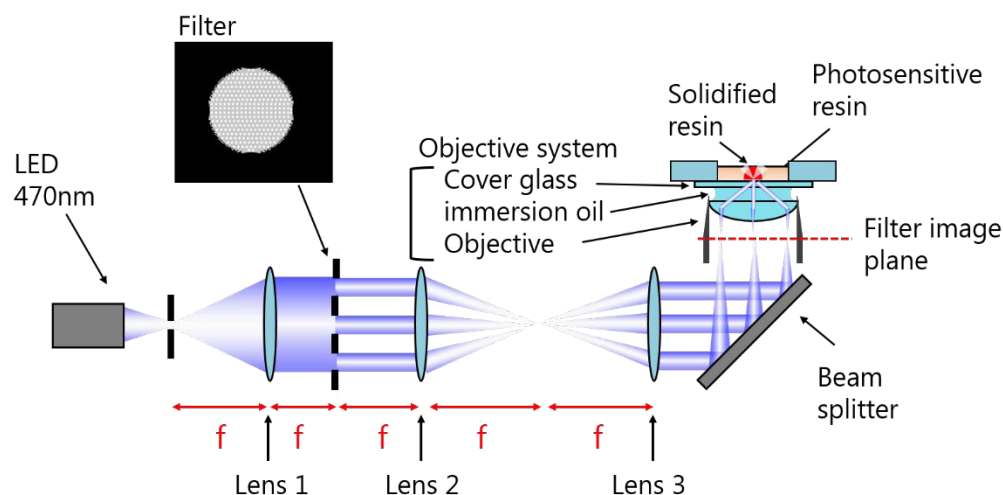


FIGURE 10. Developed fabrication system of this method, omnidirectional one-shot exposure micro-stereolithography.

CONCLUSION

In this research, in order to fabricate creatures' complex and fine structures, omnidirectional one-shot exposure micro-stereolithography was proposed. To verify feasibility, numerical simulation was conducted. It revealed that a structure like a hemisphere is able to be fabricated without filter, and a hemisphere with micro structured surface is also able to be fabricated by this method. Moreover, an exposure system was developed and experiments was conducted. As a result, a hemisphere structure with uneven surface was fabricated. The effectiveness of this method was verified.

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COMPOSITE PLATING FOR A ROLL MOLD WITH MICROSTRUCTURES ON THE SURFACE

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BACKGROUND AND OBJECTIVES

Nanoimprint technology has undergone extensive research and development as a high-throughput and inexpensive method of replicating microstructures, since its introduction [1] by S. Y. Chou, et al, 20 years ago. There are various methods of nanoimprint technology. One proposed method is roller-based nanoimprinting, in which a roller-shaped mold is used to perform continuous nanoimprinting [2]. However, fabricating microstructures on a curved surface is not easy. A proposed alternative method is to print the microstructures on a flat plate, and then to bend the plate into a roller shape. This approach suffers from the problem of having a seam on the surface of the mold.

To solve the problem, we propose a new, low-cost method of roller fabrication which produces a roller mold with no seam by using composite plating.

FABRICATING MOLDS USING COMPOSITE PLATING

Composite plating is a technology in which particles are suspended in a plating bath, and the metal and microparticles are deposited on the cathode. It is primarily used to provide the surfaces of metal materials with properties such as corrosion resistance, abrasion resistance, lubrication, and water repellency.

However, in this research we focused on composite plating's ability to create uniform curved surfaces at low cost, and its potential for use in fabricating roller mold surfaces. Our aim was to create a mold for a microlens array covering the surface of a half-sphere. The plating bath we selected for this research was a nickel sulfamate bath containing suspended SiO₂ microparticles with diameters of 1μm. We selected this plating bath because of its low film electrodeposition stress and the ease with which the bath can be managed. We selected these

microparticles because of their high hardness, non-conductiveness, chemical resistance, and thermal resistance, as well as the fact that the particles are spherical and uniform.

FABRICATION EXPERIMENT

Before fabricating the roller mold we fabricated a flat plate as a preliminary experiment. In the experiment below we used a 200 ml nickel sulfamate composite plating bath with 30 g of 1μm diameter SiO₂ particles suspended in it. For the anode we used a 10mm diameter nickel rod, and for the cathode we used a copper plate measuring 15 mm on each side.

The main factors which affected the co-deposition of the dispersed particles included the concentration of particles in the plating bath, the surface electrical charge of the particles, the conductivity of the particles, the current density of the cathode, the movement of the bath agitator and plating surface, the ion type, and the additives. The plating bath and particles to be used were decided by the target mold, so the cathode current density and bath agitation state were used as the main parameters. The impacts they had on the particles and co-deposition, etc., were tested, and the optimal conditions for mold fabrication were investigated. We determined that the optimal current density for the nickel sulfamate bath was between 2 A/dm² and 15 A/dm².

Suspended co-deposition method

Plating was performed while agitating the plating bath. Two rotation speeds of agitator rotor were tested -- 60 rpm and 300 rpm. Agitation gives kinetic energy to particles, so we expected that a higher plating speed would be needed to cause the microparticles to undergo co-deposition. Therefore, we performed plating for 30 seconds using a current density of 15 A/dm². Figure 1 (a) shows the results of plating with a rotation speed

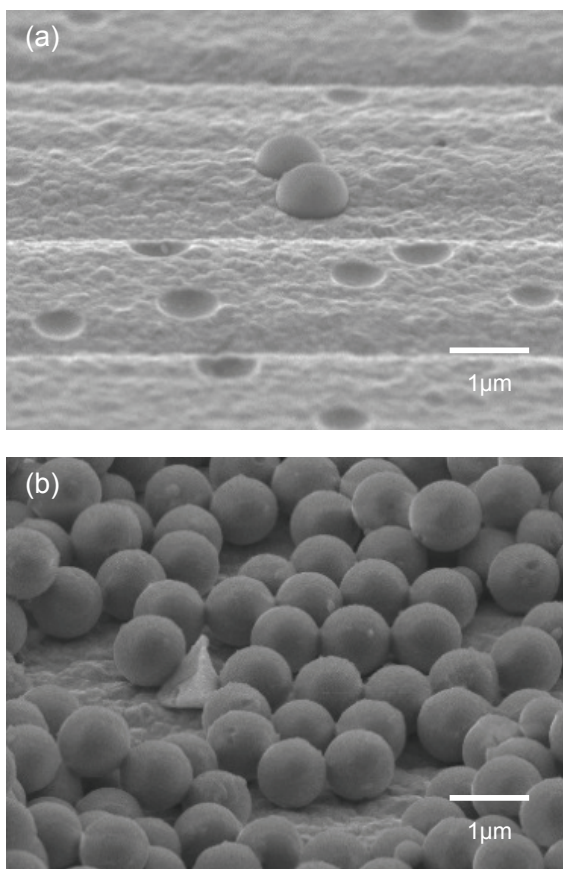


FIGURE 1. Surface After plating, (a) in stirring state, (b) in sediment co-plating.

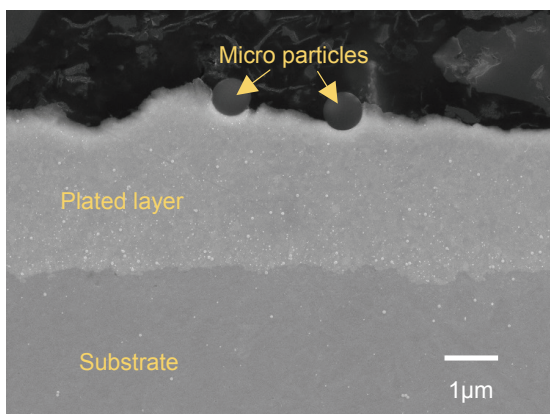


FIGURE 2. Cross-sectional view of a plating layer in sediment co-plating.

of 60 rpm. The particles were co-deposited, but the number of particles was small for a mold, and their heights were not uniform. The plating performed with a rotation speed of 300 rpm had an even smaller amount of particle co-deposition.

Sedimentation co-deposition method

Next, in order to increase the rate of particle co-deposition, we stopped the agitation and allowed the microparticles to precipitate. We then inserted a cathode in the precipitated microparticle layer and performed plating. As with the suspended co-deposition method, we performed plating for 30 seconds with a current density of 15 A/dm^2 . When we began applying current a large amount of hydrogen was generated at the cathode, so the plating was not uniform, but had a rough, unflat surface. Next we reduced the current density and performed plating for 10 minutes at a current density of 1 A/dm^2 . 1 A/dm^2 is below the appropriate current density, but the nickel was deposited without a problem, and no hydrogen was generated. The results of the experiment are shown in Figure 1(b). As the figure shows, an extremely large amount of particles were co-deposited on the surface of the cathode. However, the particle adsorption strength was low, and the particles were found to drop off merely by washing the cathode.

In order to investigate not only the surface but also the inside of the plating, we performed plating for 40 minutes with a current density of 1 A/dm^2 and examined the cross-section of the film (Figure 2). We found that no particles at all were drawn into the interior of the film. Instead, all the particles merely adhered to the exterior surface. The angle of contact between the plated metal and the microparticles was also large, preventing the particles from adhering strongly to the plating and making it easy for them to peel off.

Two step plating method

Taking the results of the previous section into consideration, in order to realize both particle adhesion and fine plated surface, we developed a method in which plating with high current density was initially performed quickly, for a short amount of time, and then, in order to correct for the insufficient plating thickness, plating with low current density was performed again in a plating bath with no particles.

In our experiment, we first used the sedimentation co-deposition method to perform plating for 10 seconds with a current density of

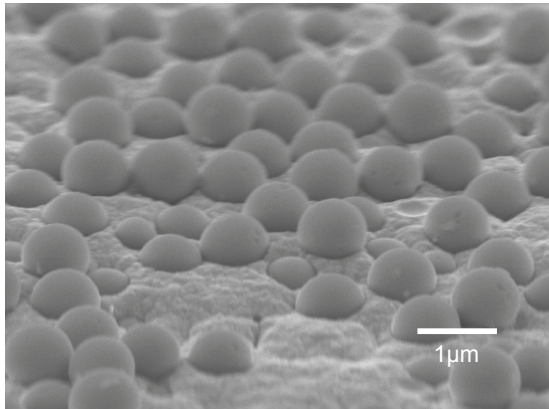


FIGURE 3. Surface after plating in two-step plating method.

15 A/dm², and then finished the plating for 20 seconds with a current density of 10 A/dm². The results are shown in Figure 3. An extremely large number of particles were securely co-deposited on the surface of the cathode, and we confirmed that when this flat plate was used as a mold to perform thermal embossing to acrylic resin film, the particles on the cathode did not drop off.

Discussions

The suspended co-deposition method resulted in a low level of particle co-deposition. We consider this is because contact between the particles and the cathode only occurred accidentally. This method is therefore inappropriate as a method of achieving high concentrations of equal thicknesses of particle deposits.

The reason for the surface roughness that resulted when using a high current density when performing plating using the sedimentation co-deposition method is considered to be due to there being insufficient Ni ions as a result of the particles being concentrated near the cathode. When a low current density was used to perform plating, the particles were not taken into the plating. We consider this is due to there being a low level of affinity between metal and the particles, so when the plating speed is low the particles drop off faster than the plating grows. It appears that with the two step plating method in the previous section the rapid plating speed causes particles to adhere tightly to the cathode, and then the finishing plating prevents the particles from dropping off.

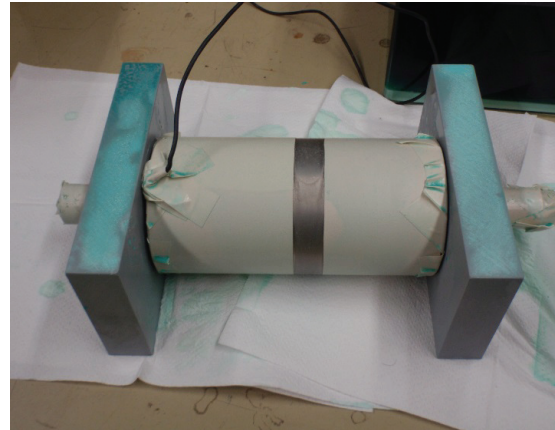


FIGURE 4. Roll mold with micro particles co-plated on its surface.

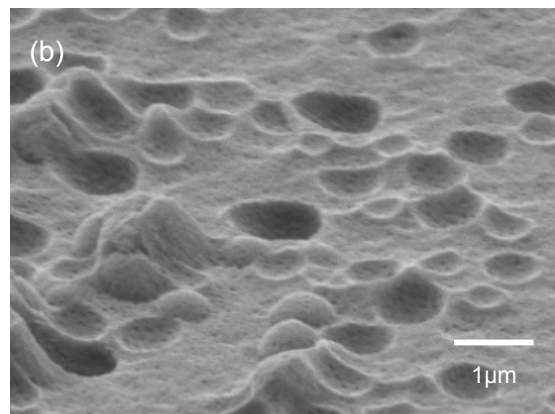
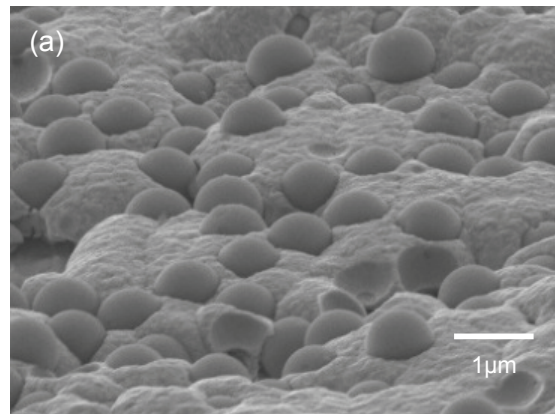


FIGURE 5. (a) Surface of roll mold. (b) Film after embossing.

EMBOSSING EXPERIMENT USING A ROLLER MOLD

The difference between plating a roller and plating a flat plate is that the roller's surface is curved. Due to the design of the device used in this experiment, the roller mold shown in Figure 4 was fabricated using a horizontal axis when inserting the cathode in the layer of particles that were precipitated. Figure 5(a) shows the results of composite plating of a rod-shaped cathode using the two step plating method. As with flat plate experiments, it was confirmed that microparticles were co-deposited.

Figure 5(b) shows the results of an embossing performed with a roller mold on resin film at a temperature of approximately 120 °C and a pressure of approximately 30 MPa. We confirmed that the mold shape was embossed onto the film surface.

CONCLUSIONS

This research project proposed methods of fabricating microstructures on the surface of a roller mold using composite plating. The results of the research were as follows.

- (1) We succeeded at high density co-deposition of microparticles on the surface of a cathode by first performing plating with high current density using the sedimentation co-deposition method, and then performing a second step of plating with low current density.
- (2) The above method was successful in creating microstructures of microparticles on the surface of a cylindrical roller mold.
- (3) The above roller mold could be used to successfully emboss microstructures onto resin film surface.

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Near – field Direct Deposition of Elastomeric Ink onto Colloidal Assembly for Preparing Large Scale Nanostructured Film with Wettability Control

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INSTRUCTIONS

Tuning hydrophobicity has offered its unsurpassed experience to researchers on design of functional surfaces with attributes, including self – cleaning, anti-icing, corrosion resistance, friction reduction and oil/water separation. Of existing fabrication methods for controlling the hydrophobicity, coating surfaces with low surface energy materials, such as fluoride and monomer with dipolar groups, were typically deployed. Alternatively, patterning technologies for fabricating nano-, micro- and hierarchical structures have shown an accelerated progress toward promoting hydrophobicity of surface. There still remain continuous needs on facile, low cost and easily scaled-up methodologies, with particularly increasing interests on establishing high flexibility which benefits wearable and wipeable applications.

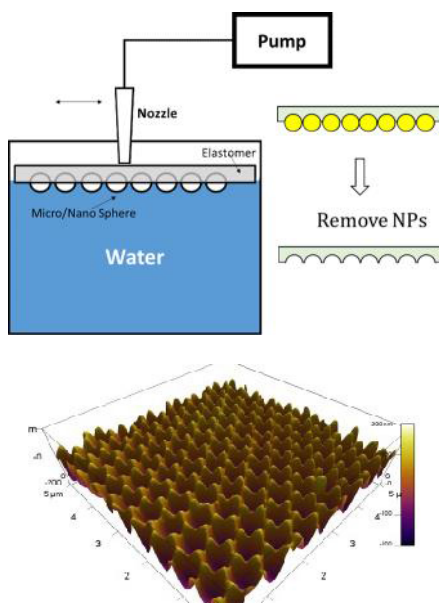


Figure 1. Proposed deposition method

In this paper, we report a design of applying a direct deposition method for fabricating a large-scale, flexible, wettability controllable, transparent and low cost nanostructured substrate. As shown in Figure 1, this new method consists of applying a thin film of elastomeric ink on the top of colloidal assembly. Well – posed process control will finally deliver nanostructured substrate which will be able to control the surface wettability.

MATERIALS AND METHODS

Hexane, sulfuric acid (H_2SO_4), sodium dodecylsulfonate (SDS) and γ -Butyrolactone (GBL) were used. Monodispersed polystyrene (PS) sphere suspensions with diameters of 500nm, 750nm and 1000nm were applied. DI water with a resistivity $18\text{M}\Omega \cdot \text{cm}$ was used in all experiments. Sylgard 184 was received and applied for preparing a PDMS solution (with weight ratio of prepolymer: curing agent = 10:1), which was further diluted with an equal amount of hexane in advance to increase the PDMS flowability.

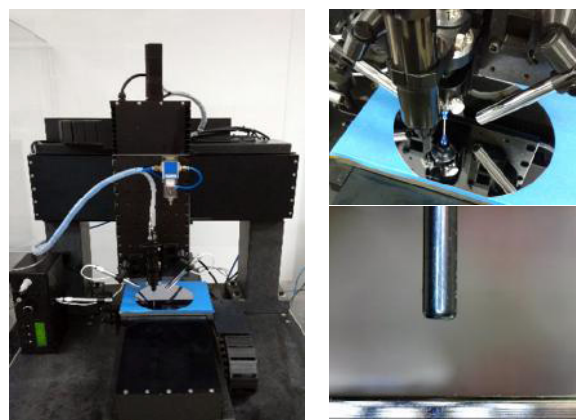


Figure 2. Prototype of Deposition System

Mono-dispersed PS NPs suspensions were diluted by mixing with an equal amount of ethanol. A compact PS NPs monolayer at the water surface was prepared through gentle dropping control. Thereafter, the prepared PDMS solution was gently applied onto the top surface of the PS NPs monolayer by using a customized 3D automated fluid dispenser system as shown in Figure 2. Since the density of PDMS solution is at $\sim 0.8 \text{ g/cm}^3$ smaller than the water density, the PDMS will float at the water – air interface for immediately wetting the PS NPs and filling the spaces among them, with a portion of the PDMS liquid film being submerged below the water level. Co-existence of triple phases among PS NPs, PDMS and water established a new interfacial balance for allowing PS rigidly and periodically embedded in the PDMS film.

The PDMS liquid film cured with a quick evaporation of hexane; and a polyethylene (PE) film, with strong affinity to PDMS, was employed for picking up the PDMS/PS film. With further rinsing by water for three time, the PDMS/PS film was sent for post – curing in an oven with different temperature settings: 60°C , 80°C , 100°C for 2 h. Then the film was immersed in GBL for 4 h to remove the PS NPs. After several cycles of washing with water and drying with N_2 gas, the nanostructured PDMS substrates were obtained.

RESULTS AND CONCLUSION

In our approach to prepare PDMS substrates with new nanostructures, the structural variations can tune the water contact angle measurements. As shown in Figure 3, for all PS microsphere templates with 3 different diameters, a contact angle (CA) of about 121° was measured on PDMS substrates with nanoring when no post-crosslinking heating treatment was employed. This value was slightly higher than that of the original PDMS film without any surface nanostructure (117°). However, when post-crosslinking heating temperature increased, nanobowl and pincushion nanostructures appeared on PDMS substrates, meanwhile the CA increased from 130° to 143° for the substrates with 500 nm PS spheres template. On the other hand, the CA decreased from 143° to 134° for PDMS substrates with pincushion nanostructures at 80°C post-crosslinking heating temperature when PS spheres diameters increased from 500 nm to 1000 nm.

However, when PDMS substrates were treated by 100°C post-crosslinking heating, honeycomb nanostructure appeared but CA decreased to 131° .

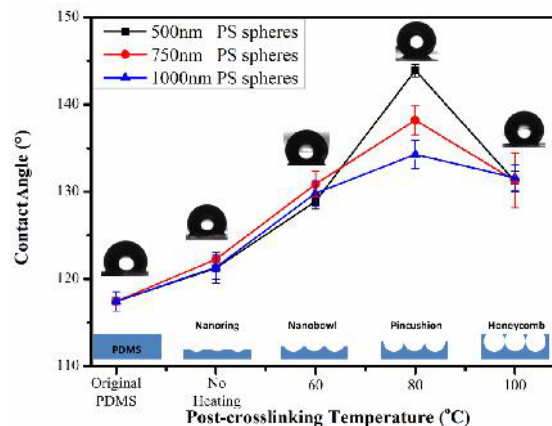


Figure 3. Wettability Control

We are herein launching a combinative method of applying direct deposition and colloidal templating method for fabricating a large-scale and flexible PDMS substrate with controllable surface hydrophobicity. Based on our unique way, the water contact angle could be increased for more than 25° , approaching to a critical WC point $\sim 150^\circ$ for superhydrophobicity. It is controlled by applied colloidal particle size and post-crosslinking. Our study demonstrated the PDMS crosslink density as a key factor in surface morphology control. The proposed method could be potential for low – cost manufacturing of hydrophobic substrate, and could be further applied in flexible sensors and actuators.

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AIR BEARING WATER-COOLED HEAT LOAD CHOPPER FOR SYNCHROTRONS

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INTRODUCTION

The Advanced Photon Source (APS) at Argonne National Laboratory is one of the largest and brightest of five US national synchrotron radiation light sources. A beam of high-energy electrons at APS emits synchrotron radiation in wavelengths suitable for imaging features of atoms, molecules, crystals, and cells. Traditional X-ray crystallographic techniques provide a static view of atomic structure. Time-resolved X-ray crystallography can provide insight into the mechanisms of molecular function. The beamline shown in Figure 1 requires precise synchronization of two rotating-disk choppers and a shutter [1]. The first chopper, a water-cooled heat load chopper, reduces the average power of the X-ray beam to protect sensitive equipment and samples from damage. The high-intensity bursts of pulses exit the heat load chopper and travel through a shutter which

isolates a single burst of synchrotron pulses. Finally, a high speed chopper is used to extract a single or multiple 100 ps X-ray pulses from the isolated burst. The timing of each chopper is synchronized with the storage-ring reference clock at 351.93 MHz.

This paper describes a heat load chopper that reduces the average power of synchrotron X-ray beams, protecting downstream equipment and samples, while improving thermal stability of the mirror alignment. The beam is chopped by a water-cooled disk spinning in a vacuum. Water is supplied to the rotating disk via an air bearing fluid rotary union. Design considerations and test results for this chopper installed at synchrotrons in the USA and France are described.

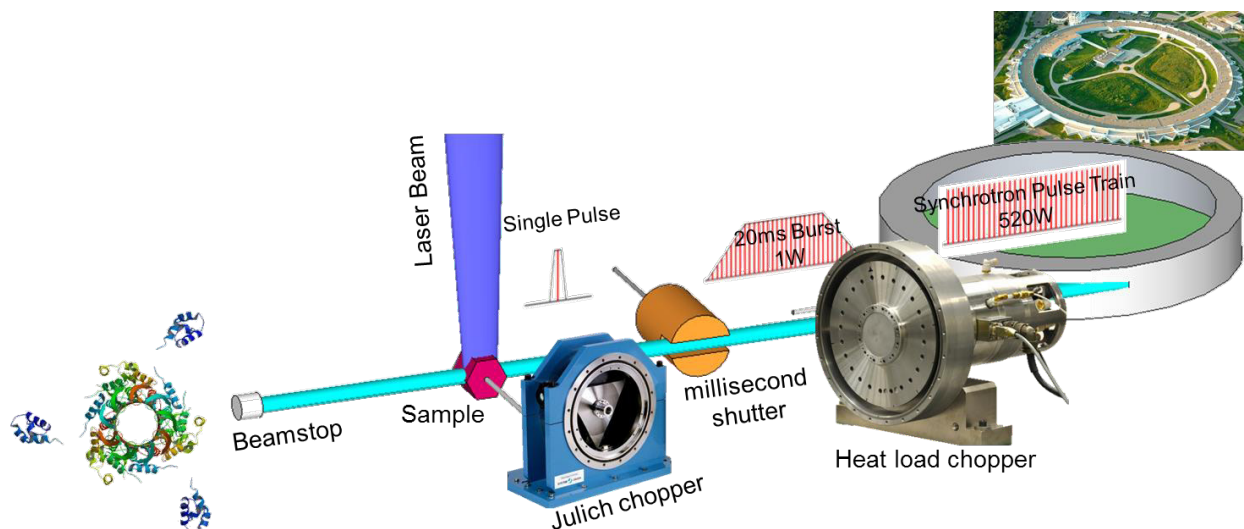


FIGURE 1. Time resolved X-ray scattering setup to provide “molecular movies” at Advanced Photon Source (APS) at Argonne National Laboratory. Photo Credit: Argonne National Laboratory.

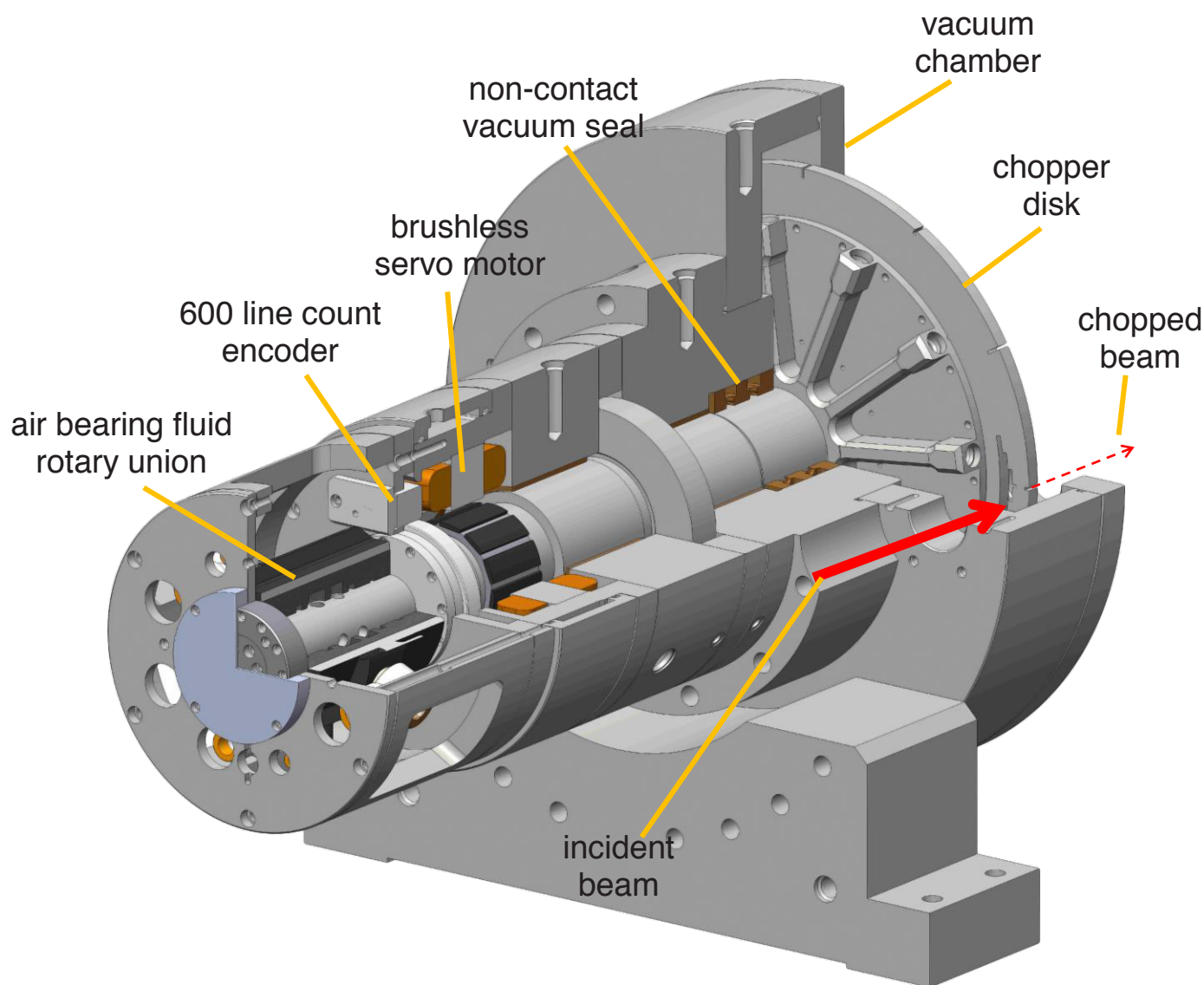


FIGURE 2. Cross-section of the water-cooled air bearing chopper spindle. The duty cycle of the high intensity X-ray beam is modulated by the rotating water-cooled chopper disk.

CHOPPER DESIGN

The heat load chopper shown in Figure 2 is designed to block a 1.5 mm diameter, 520 W X-ray beam (power density 4.7 times greater than the surface emission of the sun [1]). The beam is chopped by a slotted disk spinning in a vacuum at nominally 5 000 RPM. Vacuum levels as low as 5×10^{-6} Torr have been achieved in the vacuum chamber via differentially pumped annular vacuum seals. The vacuum seals have 10 μm gaps and provide a low friction alternative to traditional vacuum seals.

A 275 mm diameter 416 stainless disk shown in Figure 4 is 7.6 mm thick with various slot widths between 0.2 % and 1.6 % duty cycle. A variety of beam chopping configurations are available at different disk radii. As shown in Figure 3, a subslide is used to move the beam to the desired slot radius. Heat blocked by the disk is removed by coolant continuously pumped through the rotating shaft using a bespoke non-contact air bearing fluid rotary union. Non-contact air bearing seals are used to prevent leakage of the coolant without introducing unwanted error motion, jitter, or wear. These non-contact seals were originally developed by Professional Instruments for a water-cooled X-ray anode for Perkin Elmer in the 1980s [2].

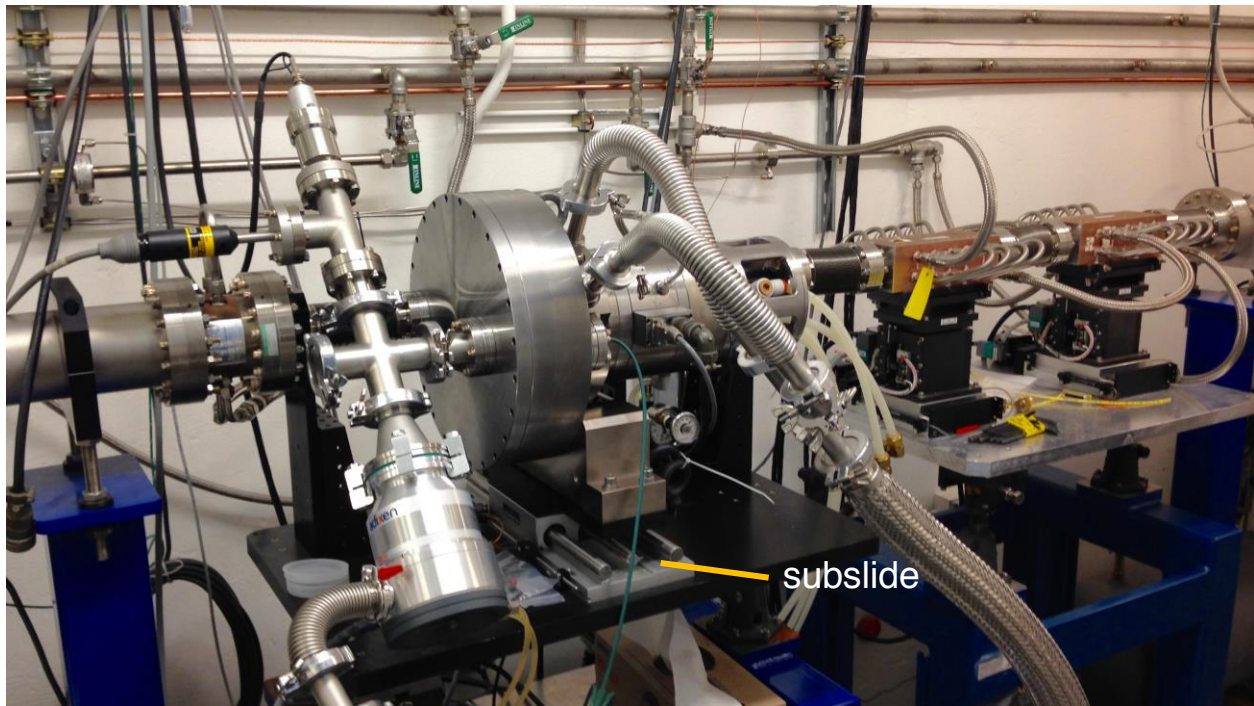


FIGURE 3. The heat load chopper installed in the beamline at APS. The subslide below the chopper is used to select the desired chopping pattern at various disk radii.

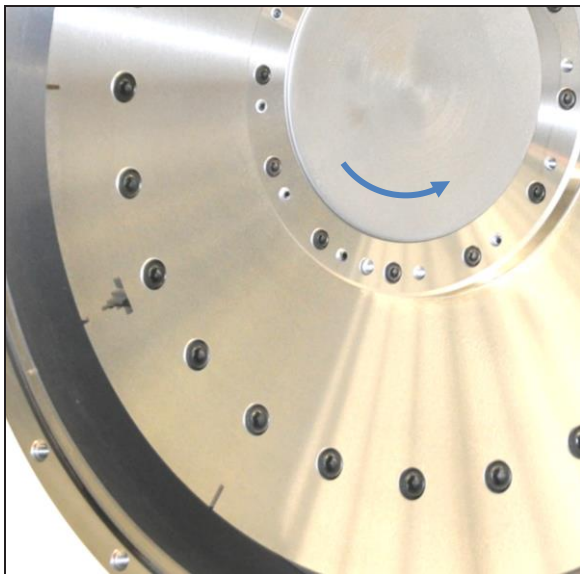


FIGURE 4. 416 stainless steel chopper disk with multiple-width slots. The chopper is positioned relative to the beam with a sub-slide to use different radii of the disk for chopping.

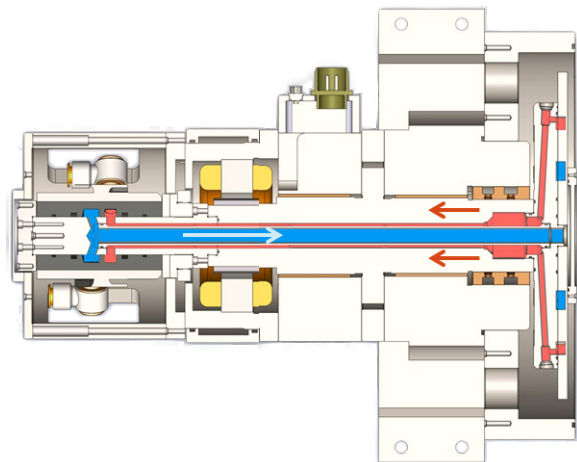


FIGURE 5. Cooling path directs coolant down the inner center channel of the spindle to the disk. Warm coolant is returned through the outer channel. The calculated temperature rise of the coolant is 2°C due to 520 W beam with a flow rate of 4 L/min.



FIGURE 6. *Picture of vacuum chamber discolored by beam backscatter. This illustrates the need for water cooling the disk that is in the direct path of the X-ray beam.*

Owing to their low error motions and torque ripple, precision air bearing spindles are the ideal choice for low-jitter choppers. The bespoke spindle used here is a captured thrust, groove compensated design with a vacuum chamber integral to the housing (Model ISO 4.5, Professional Instruments Company). A frameless brushless servo motor (MCS) and encoder (Model ERM 280, Heidenhain) is mounted directly to the rotor of the air bearing spindle. Spindle speed is controlled with a linear amplifier using Frequency Lock Velocity Control (Model LA2000, MCS). The reference signal for the control is supplied by the storage-ring reference clock enabling synchronization with the high speed chopper. The spindle is balanced with coolant flowing to the disk to better than ISO 1940 G0.1.



FIGURE 7. *Testing the flow rate of the coolant through the fluid rotary union.*

CONCLUSIONS

Design considerations and test results for a water-cooled air bearing heat load chopper with integral vacuum chamber are described. This chopper has been installed at BioCARS Sector 14 of the APS at Argonne National Laboratory where it has been operating at vacuum levels as low as 5×10^{-6} Torr. The heat load chopper is necessary as it protects downstream equipment and samples from damage from an intense X-ray beam. The heat is carried away by coolant that is continuously pumped through the rotating disk.

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DESIGN AND ANALYSIS OF A COMPLIANT JOINT FOR MITIGATING THE NONLINEAR FRICTION IN ROLLER BEARING MOTION STAGES

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BACKGROUND

Motion stages are designed for precise positioning in various manufacturing processes. Among the bearing options for motion stages, roller bearings (RBs) are the most cost effective, especially for applications requiring large motion range and vacuum compatibility. However, the presence of nonlinear friction adversely affects the precision and speed of RB guided stages [1]. The state of art in addressing this problem is to perform model-based friction compensation, which suffer from robustness problems, thus limiting their usefulness in practice [1].

Prior work done by the authors [2] has presented a simple, model-free and potentially low-cost approach to mitigate the nonlinear effect of friction in RB guided stages by the use of compliant joints. As seen from the schematic of the proposed approach (see Figure 1), each roller bearing is attached to the stage via a compliant joint. The compliance between the bearing and stage behaves as a low pass filter mitigating the effect of frictional nonlinearities on the stage. A hybrid joint design has also been proposed which combines positive stiffness flexure and negative stiffness mechanism (using buckled beam or permanent magnets). Building on the prior work, this paper investigates the realization and detailed design of a suitable compliant joint.

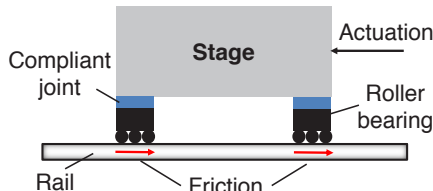


FIGURE 1: Schematic of a roller bearing guided stage with proposed compliant joints

PRELIMINARY RESULTS

Figure 2 shows a three mass model of the stage (closed loop controlled with actuation force F_a). Each compliant joint connecting the stage of mass m_s and the bearing of mass m_b is modeled

by a spring with stiffness k and a damper with viscous damping coefficient c .

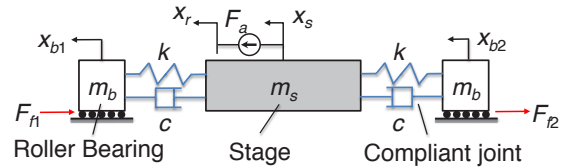


FIGURE 2: A three-mass model of the proposed stage design

Figure 3 compares the simulated closed loop frequency response functions (FRFs), from the reference position x_r to the actual position of the stage x_s , for the baseline stage, which only has roller bearings, and the compliant joint augmented stages (as shown in Figure 1), using the parameters of an in-house built nanopositioning (NP) stage by the authors [3]. In the presence of nonlinear friction F_f , there is a significant gain drop for the baseline stage between 10 – 100 Hz, compared to the case without friction, resulting reduction of bandwidth and loss of performance. Augmenting the baseline stage with compliant joints ($k = 10^5$ N/m and $c = 238$ N·s/m), the gain drop in mid-frequency is regained. If the joint stiffness is further reduced to $k = 10^3$ N/m, the FRF becomes almost identical to the case without friction, indicating the adverse effect of nonlinear friction is mitigated.

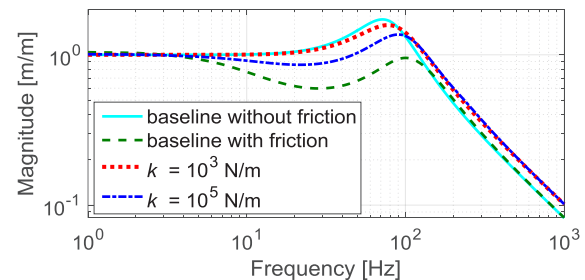


FIGURE 3: Simulated closed loop FRFs

The compliant joint augmented stage is evaluated in a case study by simulating its

settling performance in point-to-point motion. Figure 4 (a) shows the settling behavior of the system in response to a 100 nm step command (with ± 10 nm settling window). The baseline stage with friction takes 23 ms to settle, which is twice longer than the case without friction. The augmented stage with stiffness k of 10^3 and 10^5 N/m achieves 49% and 47% reduction in settling time, respectively. Figure 4 (b) shows the settling time as a function of the joint stiffness (with 3% damping ratio). As the stiffness reduces, the low-pass filtering effect of the compliant joint improves, resulting better mitigation of the nonlinear friction; note that with k below a critical stiffness (around 10^5 N/m), a significant reduction in settling time is achieved.

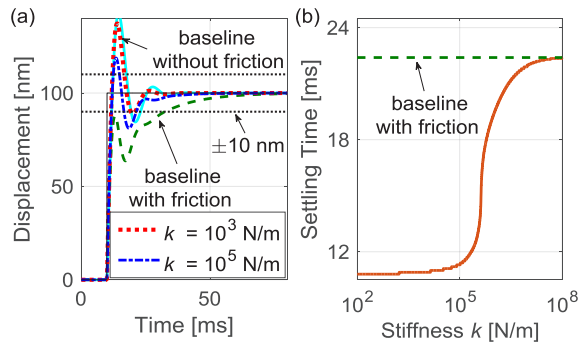


FIGURE 4. (a) Settling behavior of the proposed stage (b) effect of joint stiffness on settling time

DESIGN OF A COMPLIANT JOINT

A hybrid joint design is proposed by combining a flexure with an assist mechanism having a negative stiffness in the motion direction (i.e., axial) of the stage. The combined joint has sufficient compliance in axial direction, while remaining stiff in other orthogonal directions (i.e., vertical and lateral). The flexure is designed with leaf-springs, while a pair of permanent magnets (PMs) is used to provide the negative stiffness as shown in Figure 5.

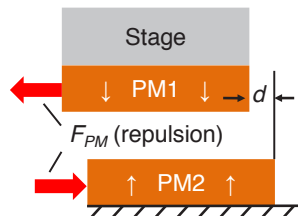


FIGURE 5. Negative stiffness with PMs

Figure 6 (a) shows the CAD drawing of the designed compliant joint, based on dimensions of the in-house built NP stage [3]; alternating poles are used in construction of PM arrays to

provide higher force densities as indicated in the figure using arrows. Figure 6 (b) shows the force-displacement relationship of the PM pair, obtained using COMSOL® finite element analysis and Table 1 summarizes the stiffness values of the flexure, PM pair, linear ball bearing and combined structure. Notice that the equivalent axial stiffness (0.02 N/ μ m) of the hybrid joint is below the critical stiffness as discussed above. Moreover, the combined stiffness of the compliant joint and ball bearing in the vertical and lateral directions have the same order of magnitude as those of the linear bearing alone, implying that the high stiffness of the bearing is not sacrificed. Results from experiments will be included in the final paper.

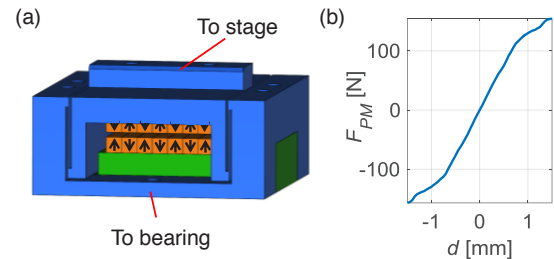


FIGURE 6. (a) CAD model of proposed compliant joint; (b) simulated force-displacement relationship of PMs

Table 1. Stiffness values of the flexure, linear ball bearing, PMs and their combination [N/ μ m]

	Axial	Vertical	Lateral
Ball bearing	N/A	175	58
Flexure	0.17	195	85
PMs	-0.15	0.21	-0.006
Combined	0.02	92	34

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MAG-6D A NEW MECHATRONICS DESIGN OF A LEVITATION STAGE WITH NANOMETER RESOLUTION

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INTRUCTION

Magnetically levitated positioning stages have been very familiar since the 90s, and extremely accurate systems are used in semiconductor industry for fine positioning of wafers. [1] [2] The main advantage in relation to “standard” air bearing systems or mechanical guided systems is that magnetic levitation provides perfect cleanroom conditions. The other advantage is that the precision of the position and trajectory motion only depends on the sensor performance and the control capability - there is no influence on inaccurate guiding - because there is no additional guiding.

Industrial applications require free access to the top plate of movable platform, which carries the probe for handling operations. Therefore, all measurement systems have to be installed on the bottom or on the side. Most of the high-resolution laser interferometers, confocal measurement systems etc. are so expensive that a system design with those types of measuring tools is possible but not relevant for certain industrial applications because of the acceptable end user price level.

We are looking for a system design that overcomes the bottleneck at the sensor arrangement and brings the price level to less than the price of standard stacked nanometer positioning systems.

The paper describes a new magnetic levitation system with nanometer resolution, a passive platform with focus on a cost-effective sensor and driver design. It enables use in several high-precision positioning tasks in 6 DOF (degrees of freedom).

The system combines a fully passive stage with 8 Halbach arrays and an optical sensor grid. The optical incremental sensor grid is placed onto the movable stage between the magnetic arrays and the coil board, which is placed on a granite baseplate. 7 optical incremental sensors heads, and additional analog hall sensors are fixed on the light-weight granite base.

MAG-6D NEW CONCEPT

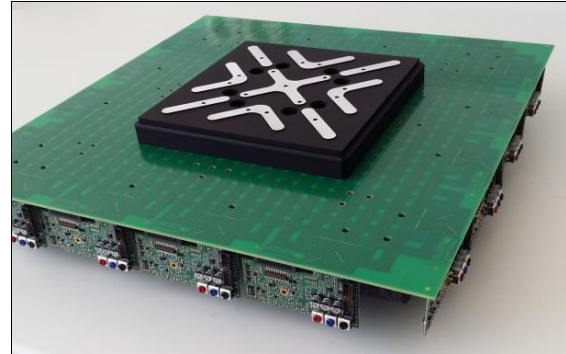


FIGURE 1: MAG_6D without cover

Figure 1 shows the MAG_6D system without cover, the power and sensor-data cables. The passive stage carries the Halbach arrays and a 2-D grid. All drivers are mounted in connectors directly on the PCB.

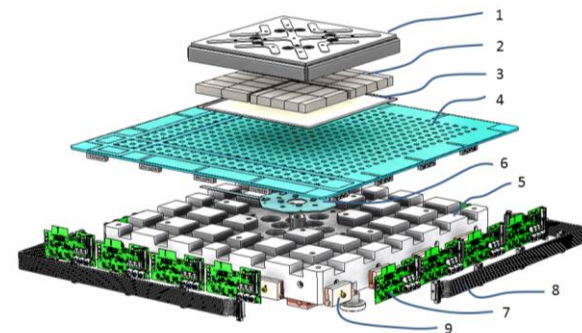


FIGURE 2: Assembly of the MAG_6D

- 1 Movable stage
- 2 Halbach array (consist of 8 arrays)
- 3 2D optical grid
- 4 Coil board (PCB with 144 coils)
- 5 Granite baseplate (nonmagnetic)
- 6 Sensor module with 7 sensor heads
- 7 Current drivers (48 full bridges)
- 8 Cable set-up and sensor connector modules
- 9 Heat pipe for very low thermal hot spots

8 Halbach arrays are placed in two pairs on each corner and rectangularly in two pairs.

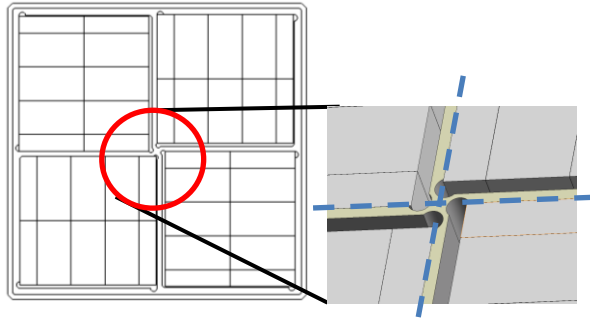


FIGURE 3: View of the bottom side of the stage (the 2-D optical grid is removed)

2-D OPTICAL GRID

Metal bridges with a thickness of a few millimeters are placed around each Halbach array pair to support the 2-D optical grid. The 2-D optical grid consists of a hardened glass substrate with chromium dots and has a thickness of 0.7mm. This special glass substrate together with the suspended metal bridges keeps the grid surface reference flatness at less than 4 μm ripple. The surface will be mapped into the Z sensor control matrix with about 2000 points. In our formal magnetic levitation design [4], the magnetic structure was mapped into about 10000 data points and was also used for on-the-fly error correction in the control algorithm.

HALBACH ARRAYS WITH LARGE PERIODS

The force calculation is done with a combination of FEM magnetic field methods, a Lua script based on Python and a newly designed, fully graphical software interface. Fig.4 shows the good correspondence of the calculated magnetic field components to the measured field distribution. Symmetrically Halbach arrays are chosen for generation of the strong magnetic field. To reduce the weight of the movable stage, the thickness of the Halbach arrays was halved. That reduces the weight by a factor of two but reduces the field in the board layer only by a factor of 1.3. The 3-D field magnetic measuring technology based on an “F3A Magnetic Transducer” - a 3-D magnetic Hall probe - from SENIS. [3]. For the scan operation, our high-precision positioning technology synchronized with PI’s hexapod controllers and a proprietary GUI program tool is used.

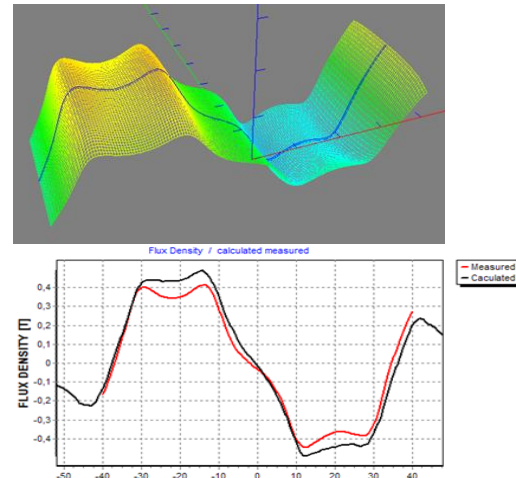


FIGURE 4:

One segment of the 8 Halbach arrays, period of 42 mm a) 3-D - measured magnetic flux density in a horizontal direction (X); the blue line shows the cross section for 2-D b) 2-D - verification between measurement and simulation shows acceptable conformity

COIL DESIGN

Coil design with mechanical gaps between the electrical phases. These gaps are used for fixation of the board and for accessing the sensor heads through the board onto the 2-D optical grid.

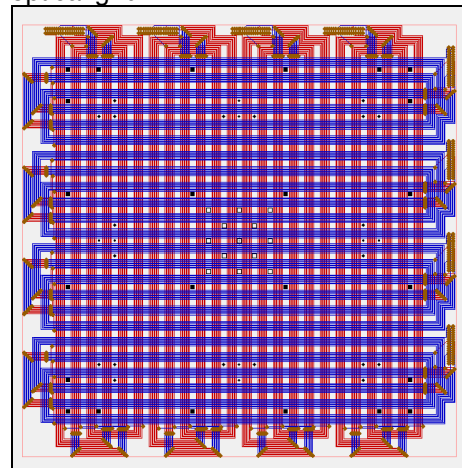


FIGURE 5: One layer of the PCB with the 144 coils

The coils are placed rectangularly onto the upper and lower side of this single layer, buried vias are used to allow the cross section of the coil structure. Each horizontal and vertical coil is connected to a connection area where the connecting point shifts in each layer to the next pin section on the current driver connector.

The coils inside the PCB are placed at gaps of 5 mm. Some of the gaps have rectangular holes through the PCB to open the light path for the incremental optical sensor heads. The stage design is optimized for high electromagnetic forces. The static lifting force (Z direction) is more than 4 times the weight of the stage and can overcome 6 times this value for dynamic operations. This is much higher than in our former design with the six-coil structure. [4]

With a moderate current of only 1.5A in the clusters of the coil assembly, the stage can be driven by more than 40 N in a vertical and horizontal direction. The coil structure (in 12 layers on a PCB) does not need additional wiring and is connected with highly efficient PWM drivers, placed directly on the main board. The drivers are controlled by digital serial SPI channels – the digital connection to the controller.

The 48 full bridge drivers have a very efficient design and this reduces the risk of EMI distortions, because of the short connection to the board, the high PWM frequency, and a simple filter function. The PWM frequency is set to 400 kHz and is synchronized between the boards.

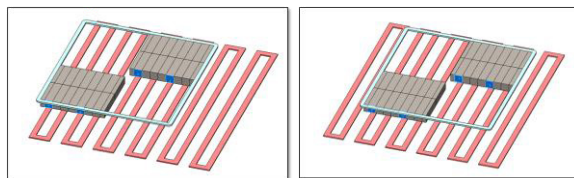


FIGURE 6: Tolerances on Halbach arrays have no influence on the forces generated at different X-Y positions. (Was already discussed in earlier paper by X.Lu et al.)

Maglev systems with PCB coils are already familiar in the literature, but these systems need measuring sensors which span over the upper working surface of the movable stage. In that case, the stage can't be used in industrial applications where the upper surface has to be free for the customer scanning application. [5] In our new design, the platform surface is free. The coils inside the PCB are placed at gaps of 5 mm. We could also see that the efficiency of the Halbach array design is better than in a configuration with three long Halbach elements.

Some of the gaps in the coil board have rectangular or cylinder holes through the PCB to open the light path for the incremental optical sensor heads and for additional hall sensors for the initialization procedure. The system uses a set of 4 incremental sensor heads separated by 28 mm, placed in 4 of the small gaps on the main PCB board. The small sensor heads are assembled on 4mm rectangular piggy-back boards to get the designated height. The sensor noise was determined to less than 8nm (RMS).

GRANIT BASE PLATE

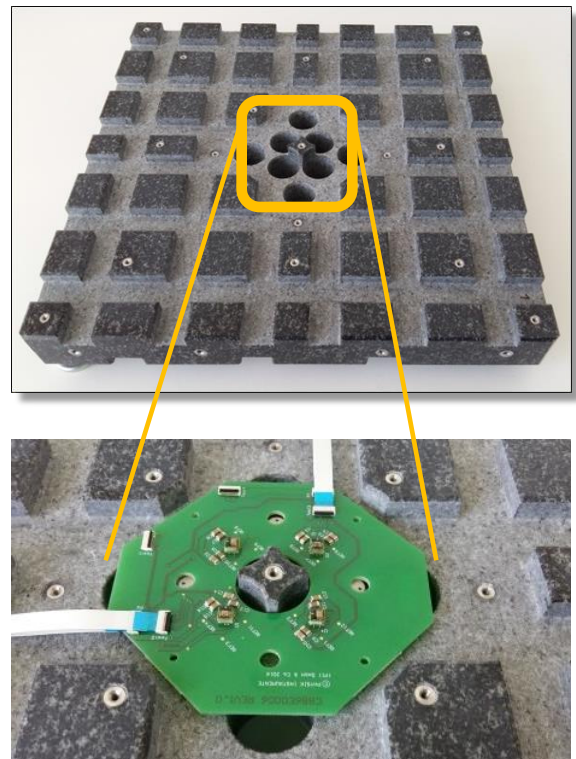


FIGURE 7: implementation of the 7-head sensor module, 4 optical incremental sensor heads and 3 interferometer distance heads

Nonmagnetic granite was chosen for the baseplate. The PCB and the sensor modules are fixed in nonmagnetic inserts on the granite plate. Thermal conductive silicon is used between the PCB and the granite base plate; thermal conductive graphite was also tested. That layer of thermal conductive material reduces formation of ununique thermal zones on the PCB.

TEST RESULTS

To check the performance of the magnetic structure, the 3 phase coils and the drivers, a number of magnetic distribution measurements are done with high-resolution 3-D magnetic sensor chips from Austria Microsystems.

The following measurement results show high consistency between the desired magnetic distribution of the flux density on the coil level between the computer simulation and measurement. It is an overall test including controller functions.

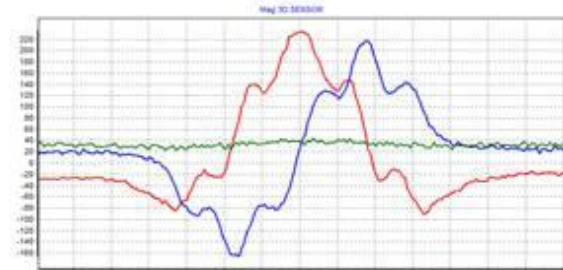


FIGURE 8a: Measurement of flux density 2 mm over the coil level (PCB) for "3-phase lifting current condition"

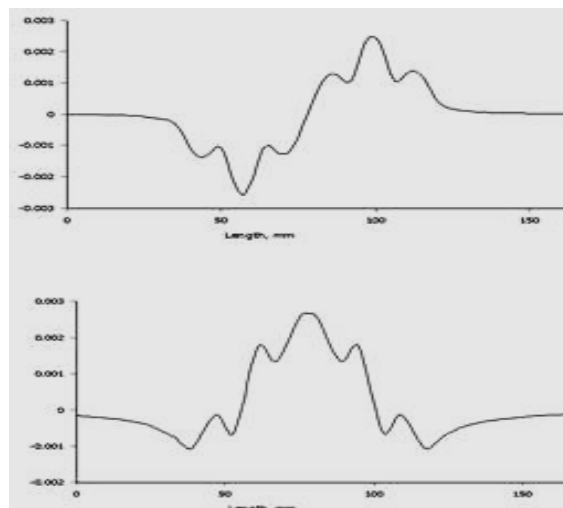


FIGURE 8b: Computed flux density in normal and tangential direction, generated by the 3-phase coil current.

Because of the high PWM frequency of 400 kHz, we could use $6.8\mu\text{H}$ chokes with small dimensions for the low-pass filter function. The PWM ripple in the coil layer is smaller than 100mV.

DECOUPLING OF LIFTING AND X/Y FORCES

There is a 60-degree phase angle of spatial frequency between phase1_2 and between phase 2_3.

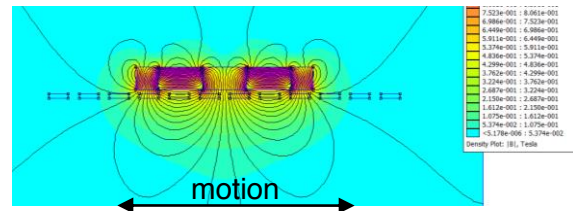


FIGURE 9: Flux simulation of one Halbach array

The spatial frequency current distribution for three phases of the coil structure can be defined as:

$$\text{ampl} := 1 \quad (1)$$

$$\text{phase1}(x) := \text{ampl} \cdot \cos\left(\pi \cdot \frac{x}{42} - \frac{\pi}{3}\right) \quad (2)$$

$$\text{phase2}(x) := \text{ampl} \cdot \cos\left(\pi \cdot \frac{x}{42}\right) \quad (3)$$

$$\text{phase3}(x) := \text{ampl} \cdot \cos\left(\pi \cdot \frac{x}{42} + \frac{\pi}{3}\right) \quad (4)$$

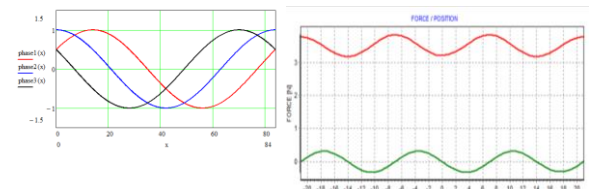


FIGURE 10a: Shows a 10% sinusoidal distortion of the lifting force and of the horizontal force component. (spatial frequencies)

$$\text{ampl}(x) := 1 + 0.09 \cdot \cos\left(\pi \cdot \frac{x}{7}\right) \quad (5)$$

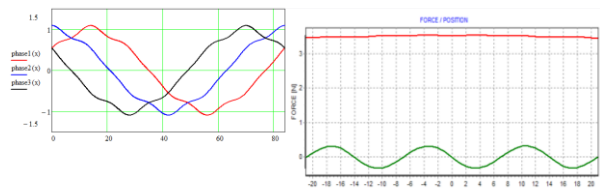


FIGURE 10b: The spatial frequency amplitude modulation (5) results in decoupling of the lifting force without influencing the other force component.

It can be shown that the horizontal component can be also decoupled and linearized by using a spatial frequency phase shift between two phases. With this feed forward term, the control function can be linearized despite the fact that the gaps in the coil structure distort the linear properties.

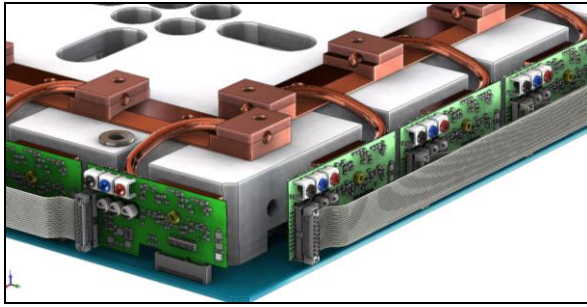


FIGURE 11: Bottom view of the stage, heat pipes between the current driver modules and the heat sinks on the granite base plate

Because the coil drivers are designed with highly effective PWM full bridges, there is moderate heat generation even without an air cooling stream. The heat generation is further reduced by using passive heat pipes.

CONTROLLER STRUCTURE

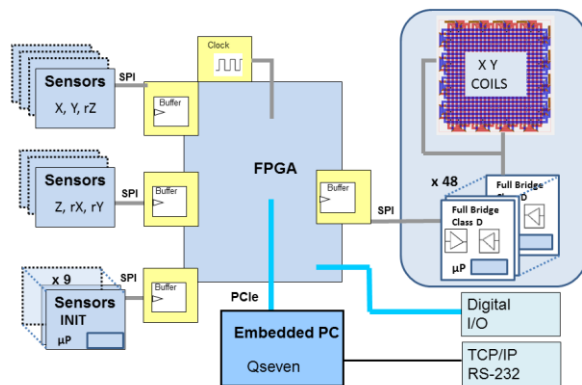


FIGURE 12: Controller structure

We chose a well-known structure for the controller.

- Digital channels for the 7 sensors + 9 ratiometric Hall sensors
- Driver close to the stage
- Power direct to the driver
- Digital-controlled channels (24) to the driver
- FPGA with crystal clock for all sensor interfaces
- Intel's PC processors (4 core) for high computing power, coupled with PCI express lanes and direct memory access
- Real-time operating system on the PC platform, 20kHz sample clock
- Option for 7" touch display

This design also opens the way for using the complete PI Software package. Command structure and trace functionality are available.

The control commands are the same as those used for a 6-DOF piezo stage.

FURTHER WORK

At the time of publication, some components were not available in industrial product quality, so the controller consists of the target components but is not at the quality level of an industrial-proven model ready for shipping. We also made tests with different low-budget, high-resolution interferometric distance sensors for the short displacement of 4 mm using the same digital interface. Generally, the robustness of the new system has to be qualified and it is possible that particularly the Z sensors may need to be modified.

Although there is still some room for improvement of the magnetic field design and the control performance of the magnetic levitation system, the real challenge is choosing the right sensor systems for a low-budget mechatronics system.

SUMMARY

A new magnetic levitation system could be realized with a PCB-based coil structure with gaps between the phases and a 2-D grid that span over the bottom side of the movable stage. The new and very compact low-budget design of the MAG-6D system is scalable. This scalability issue allows adaption of the stage design to a number of planar positioning applications in industry and research.

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PRECISION PIPETTE MANIPULATION FOR AUTOMATED, SERIAL NEURAL RECORDINGS *IN VIVO*

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INTRODUCTION

Whole-cell patch clamping [1,2] is the gold-standard measurement technique for recording the electrical activity of individual neurons in the brain, and is used by thousands of labs worldwide. This technique is able to record the response of individual neurons to sensory inputs to the brain (e.g., touch, sight) and yields direct insight into how the brain processes information, forms connections and what goes awry in disease. While useful, the technique is generally considered an art form and requires years of training to become proficient. It is a highly repetitive manual process requiring patience and intuition to efficiently gather recordings. We report the design and performance of an automated system for performing repeated, serial recordings in the living brain (*in vivo*), focusing critically on the precision manipulation of the pipette that is used for the recording interface to neurons in the brain.

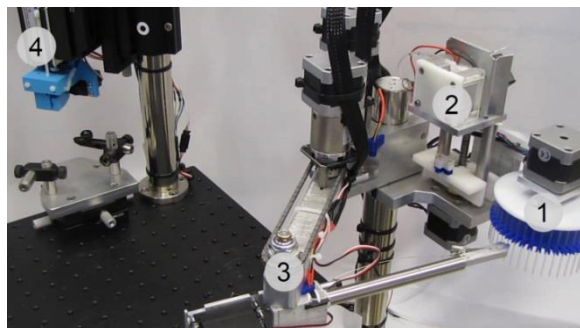


FIGURE 1. (1) Pipette storage carousel. (2) Pipette filling station. (3) 2 degree-of-freedom robot arm. (4) Recording headstage with v-groove kinematic constraint. Mouse (not shown) is positioned below the constraint shown in (4).

NEURAL RECORDING AUTOMATION

We have developed a robot (Fig. 1) that has performed, for the first time *in vivo*, multiple serial patch clamp recordings without manual

human intervention. In operation, pipettes used for recording are delivered to the recording headstage (Fig 1-4) by a two-link planar robot arm. The arm picks and places a requisite glass recording pipette (1.5 mm diameter, 53 mm long, last 3 mm tapered to 1 μ m diameter tip) from a storage carousel to a fluid filling station, and subsequently to a kinematic constraint on the recording headstage. After kinematic constraint, the pipette is lowered into the brain for recording. After recording, the pipette is automatically removed and replaced to repeat the process. This precise pipette manipulation system frees experimenters from manually performing the 4-6 hr experiments and enables new types of recordings, such as measuring sleep patterns, where the presence of the experimenter may disturb the animal.

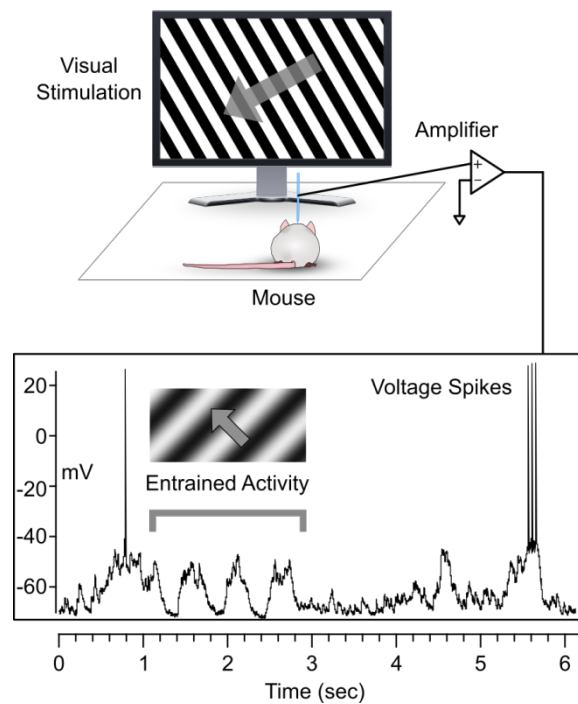


FIGURE 2. Top: schematic of the experimental setup to record the voltage response of neurons in the visual cortex of the brain. We display drifting sinusoidal gratings of differing spatial and temporal frequencies to stimulate the cells and elicit a response. Bottom: Example recording with intracellular sub-threshold voltage response to visual stimuli and spontaneous voltage spikes.

The robot has recorded “hands free” from over 40 neurons and performs with similar quality and throughput to that of a highly skilled manual operator, but without manual intervention. Figure 2 shows an experimental setup with an anesthetized mouse located in front of an LCD display that shows translating black and white sinusoidal gratings while simultaneously recording the electrical response from a neuron in the visual cortex. These experiments are designed to elucidate the different types of neurons in the visual cortex and their roles in processing visual information.

PRECISION PIPETTE HANDLING

The robot arm can position the pipette repeatably in a 450 mm radial workspace shown in Fig. 3. The arms are driven by planetary gearboxes with 99.51:1 reduction attached to stepper motors with 1.8° per step. The full step resolution of the robot is 0.0181°/step with a linear resolution of 144 μm /step at the end effector. Microstepping was utilized to reduce motor noise and increase smoothness. A simple custom embedded controller allows simultaneous motor control with variable step rates. The path planning for the robot is performed graphically using a vector graphic drawing program that, when combined with reverse displacement kinematic analysis (Fig. 3 bottom), exports the desired paths as angular positions in embedded firmware for the controller. The repeatability of the robot is better than $\pm 150 \mu\text{m}$, sufficient for transferring the 1.5 mm diameter pipettes between stations.

A critical design feature of the robot arm, and all of the stations (Fig 1), is having sufficient flexibility and precision to accurately position the pipettes without breaking them. At several stations, compliant receptacles enable pipette docking accurately and repeatably without fracturing the glass. The receptacles are 3D printed pairs of vertically oriented, symmetrical cantilevers (0.6 mm x 5.4 mm x 5.7 mm, ABS plastic) oriented so the pipette is retained

vertically by friction between them. They are designed to be sufficiently flexible to avoid pipette breakage, while maintaining positional accuracy by passively centering the pipette between them. These receptacles were used in the storage carousel and the filling station.

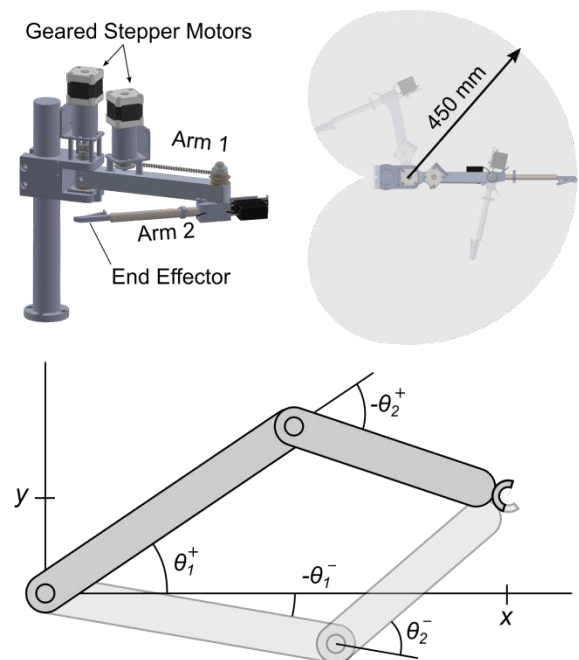


FIGURE 3. CAD rendering of the 2 DOF robot arm, 450 mm radial workspace, and the variables used in the reverse displacement kinematic analysis.

PRECISION PIPETTE TIP PLACEMENT

One of the largest contributors to success in a patch clamp recording experiment is the health of the tissue under investigation. For *in vivo* experiments, this requires the surgical preparations to be pristine, and the tissue to be extremely mechanically stable. The 1 μm diameter tip of the pipette must rest against the membrane of a 10 μm diameter neuron for the duration of the recording. The stability of this connection is acutely affected by motion in the tissue due to breathing and heartbeat of the animal. One of the ways to minimize the effect is to reduce the size of the hole made in the skull to access the brain. For maximum stability, a craniotomy of approximately 200 μm in diameter is optimal. However, the useable area within that diameter is often smaller due to presence of remaining bone debris, the dura (membrane surrounding the brain), and blood vessels within the tissue. Figure 4 shows a schematic of these common obstacles for a

more typical 750 μm craniotomy. The dashed circle in the figure represents the desired 60 μm diameter target area where the tips of the pipette should be placed.

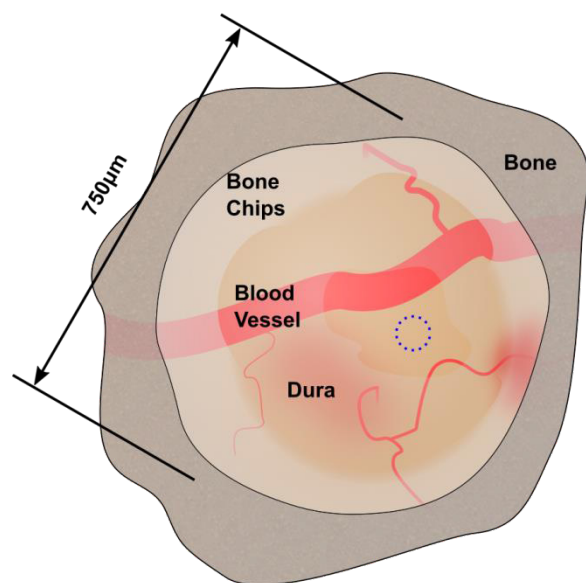


FIGURE 4. Representation of a craniotomy in the skull of a mouse and the exposed brain tissue. The obstacles within the craniotomy including bone chips around the internal periphery, the dura membrane, and blood vessels. The dashed circle (60 μm in diameter) is the desired tip location for the pipette and any position errors could result in a broken pipette and/or tissue damage.

To achieve this placement accuracy, we designed a vertically oriented v-groove constraint to repeatably position the walls of the pipette to within ± 30 μm . Figure 5 shows the positional repeatability and accuracy from repeated measurements of multiple pipettes inserted into the v-groove constraint in a random order. It shows that individual pipettes are positioned repeatably (error of $\sigma = 2.8$ μm) but the overall repeatability is worse $\sigma = 15.4$ μm for the group.

The pipette-to-pipette variation is likely caused by slight out-of-round errors (max OD minus min OD ≈ 10 μm) in the pipette itself giving the cross section of the pipette an elliptical shape. There is also some variation in their average diameters (1.48 mm to 1.56 mm) which can cause an error of up to 56 μm in the axis passing through the apex of the V.

Previous versions of the pipette constraint used tapered collets similar to those used in CNC milling machines which, while they are less sensitive to diametrical variation, were eventually rejected due to manufacturing costs, interactions with the recording amplifier, and Abbe error along the length of the pipette. The radial repeatability error of this constraint was $\sigma = 48.8$ μm . Both designs successfully recorded from neurons *in vivo* but the collet design had much higher rates of pipette clogging because the tip came in contact with the dura. After decreasing the position error in the constraint by 68.5%, the clogging rate was reduced by 80%. This significantly increases the yield of the robot and enables much higher throughput collecting much larger data sets.

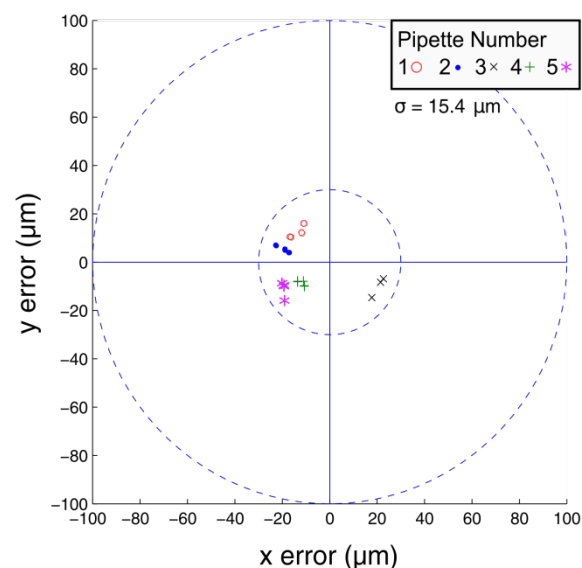


FIGURE 5. The outer dashed outer circle represents the diameter of the a more stable craniotomy and the inner dashed circle represents the desired 60 μm diameter target area. The markers are measurements of the positional repeatability of pipette in the v-groove constraint.

We also characterized the error between the tip of the pipette and the walls of the pipette. To manufacture a pipette, a borosilicate capillary is threaded through a circular platinum filament which heats a 3 mm section in the center of the capillary. The capillary is then pulled from both ends and the softened glass near the filament necks down and breaks to produce the 1 μm glass tip. There is some random error in the bearings of the puller mechanism and in the

filament position which leads to bending of the pipette tip off of the central axis of the pipette. We measured this error by placing 4 microscopes in an opposing arrangement to simultaneously measure the position of the center axis and the tip of the pipette. Figure 6 shows the microscope arrangement, Figure 7 shows the image processing steps (central axis in red), and Figure 8 shows the position error between the tip and central axis.

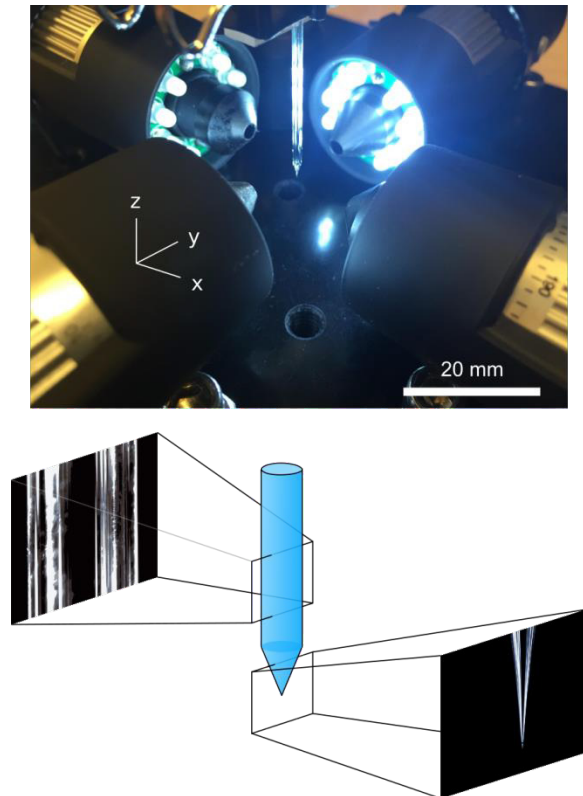


Figure 6. Top: opposing arrangement of four microscopes to measure the position of the central axis of the pipette simultaneously with the tip in both the X and Y directions. Bottom: the view from a pair of microscopes showing the simultaneous measurement.

This manufacturing error was found to be $\sigma=29.5 \mu\text{m}$. The combination of the manufacturing and v-groove repeatability error give the overall tip positional repeatability of the system as $\sigma=33.3 \mu\text{m}$, similar to other reports [6]. While this is sufficient to meet the $200 \mu\text{m}$ diameter craniotomy size and was very successful operating autonomously *in vivo*, the yield of the robot could be further improved by reducing this error. The quality loss function, or in our case the precision verses performance

function, pioneered by Taguchi [7] is well supported by the relationships observed during *in vivo* experimentation with the robot. The total error measured is approximately two times the desired error for a 95% confidence interval. Given the random error in manufacturing, we believe the solution is to provide an on-line non-contact direct measurement of the tip position after insertion into the v-groove constraint with better than $10 \mu\text{m}$ resolution. This is complicated by the limited physical space, tight electromagnetic interference requirements, and limited travel range of the actuators that position the pipette in the brain. However, by using feedback the accuracy could be substantially improved over the current open-loop precision implementation.

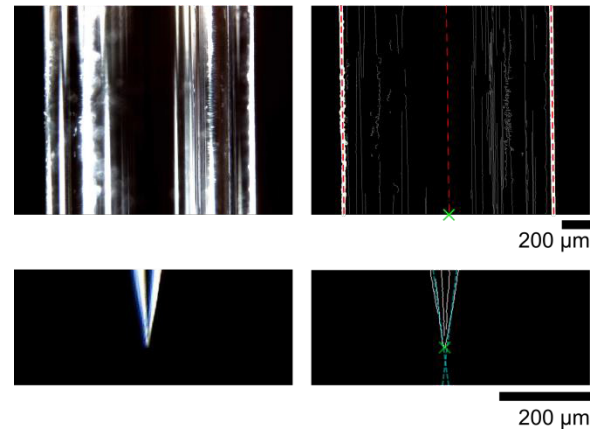


Figure 7. Left: raw images. Right: images where edge detection, and linear regression was performed to find the central axis of the pipette (red) and the location of the tip of the pipette (green X, bottom right).

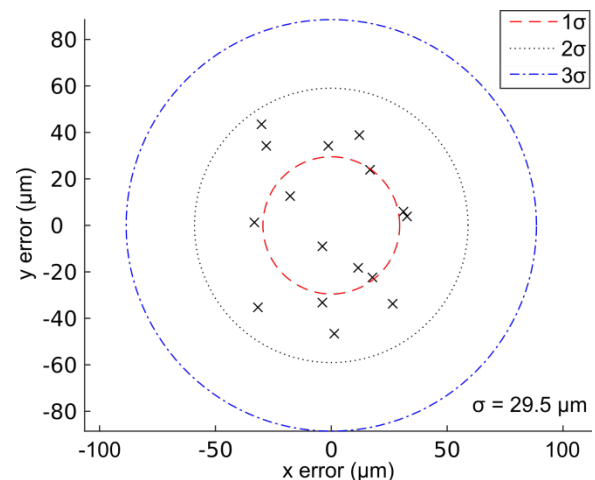


Figure 8. Off-axis error between the walls of the pipette and the 1 μm diameter tip due to manufacturing variation.

RECORDING STABILITY

Even more important than pipette tip positioning accuracy, which can affect the yield of the robot, is the stability of a recording after a pipette is successfully inserted into the brain. Over a 60 min recording, the delicate connection between the pipette tip and the cell can be disrupted if there is drift in the position of the pipette of more than approximately 4 μm . Drift can cause the membrane of the cell to tear or biological debris can clog the tip of the pipette, both of which reduce the signal to noise ratio of the recording or can completely dislodge the recording. Sources of drift are the heating cycles of the warming pad under the or from high power usage components such as motors or the LCD monitor that displays visual stimuli.

We characterized the drift of an initial prototype and discovered that the multitude of cables, motors, and cantilevered masses, combined with the thermal sources, contributed to vibration and mechanical drift. Using a capacitive displacement sensor, we measured the drift to be approximately 3 $\mu\text{m/hr}$ which is acceptable, but is approaching the limit required for stable recordings. The target drift for most systems used in electrophysiology is < 1 $\mu\text{m/hr}$. To achieve stable recordings, we redesigned the system to isolate the structural loop of the pipette from the rest of the automation hardware as shown in Fig. 1. The robot arm was added to transfer pipettes between these structural loops. The measured drift of the pipette in the new architecture is 0.6 $\mu\text{m/hr}$ which dramatically increased the average recording duration from 4.1 to 20 minutes and the maximum recording duration from 8.8 to 110 minutes. We have employed this increased performance for automated, high throughput measurements of neurons in the visual cortex examining their basic functional properties and developing the basic cell-type classification schemes based on their roles in the visual processing pathway.

CONCLUSIONS

Using precision machine design principles of kinematic constraint, structural loop minimization we have characterized and removed many mechanical errors (thermal drift, pipette position accuracy and repeatability) that make automation of patch clamp recording

challenging. As a result, we have developed a robotic system that, automates serial patch clamp recordings for *in vivo* intracellular electrophysiology. The precision improvements have significantly increased the throughput, yield, and quality of automated intracellular electrophysiological experiments and enable parallelization and enhance repeatability of the technique.

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Development of a micro-turbine utilized an extra-fine wire

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INSTRUCTIONS

In 1980's, micro machines (MEMS) and those micro mechanical parts have been developed. A micro gear, a micro screw, or a micro linkage was produced as early trial products. These micro parts were made by photolithography technique, by LIGA process, or ultra-precise machining. After that, a rotary micro motor, a comb drive actuator, and a micro turbine[1] were developed. These micro actuators were driven by electromagnetic force, by electrostatic force, or by air jet. However, the micro parts and the micro actuators have not been combined as the micro machines to do useful work, except for a few researches. Although micro sensors and micro mirror array were developed, these micro systems could not take out strong force or big motion.

In this paper, a micro-turbine impeller is developed by utilizing an extra-fine wire. The micro-turbine impeller is made by binding, hooking, and knitting the extra-fine wire around a small disc and its shaft. At the same time, the spur gear is also made on circumference of the disk. In this micro turbine, the micro gear is a micro-mechanical element, and the micro-turbine impeller is a micro actuator. Therefore, both components are combined as a micro machine to do useful work. Furthermore, the micro gear integrated into the micro-turbine impeller is engaged with outside gears, and this micro turbine system can take out strong force and big motion.

In this paper, structure and making processes of the micro-turbine impeller are described, and rotational performances of the micro turbine are confirmed by experiments.

STRUCTURE OF THE MICRO-TURBINE IMPELLER

The micro turbine impeller consists of an extra-fine wire, a small disc, and its fine shaft as shown in Figure 1. The extra-fine wire is bound, is hooked, and is knitted around the small disc and its fine shaft. Knitted extra-fine wire forms the micro-turbine blades on both sides of the disc, and arranged extra-fine wire composes a spur gear on circumference of the disk. When

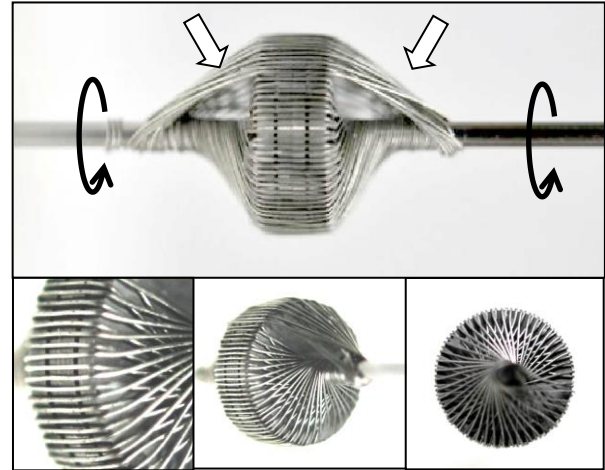


FIGURE 1. The micro-turbine impeller utilizing an extra fine wire

TABLE 1. Specification of the micro-turbine impeller, the spur gear, and the extra fine wire

Diameter of the turbine	: 11.66 mm
Length of the turbine	: 19.8 mm
Weight of the turbine	: 3.65 g
Number of the impeller	: 2 wings
Diameter of the shaft	: 1.6 mm
Length of the shaft	: 50.0 mm
Module of the gear	: $m=0.1273$ mm
Numbers of teeth	: 90 teeth
Pitch diameter	: 11.46 mm
Face width	: 4.0 mm
Diameter of the wire	: 0.2 mm
Wire material	: Copper wire coated of tin
Diameter of the disc	: 11.26 mm
Width of the disc	: 4.0 mm

the air jet blows from the directions of white arrows in Figure 1, the micro-turbine impeller rotates to the direction of black arrows. Table 1 shows the specification of the micro turbine impeller, the spur gear, and the extra-fine wire.

Advantages of the micro-turbine are simple structure and simple manufacturing process in which does not use expensive and large-scale equipment. Additionally, the micro-turbine can rotate without electrostatic breakdown.

MAKING PROCESS OF THE MICRO-TURBINE IMPELLER

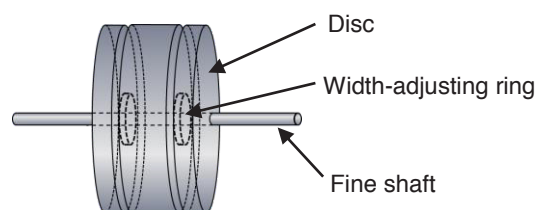
Figure 2 shows the making process of the micro-turbine impeller utilizing an extra-fine wire.

Firstly, a winding barrel is composed by three discs, two width-adjusting rings, and a fine shaft. The fine shaft is inserted at center holes of the discs. The width-adjusting rings are arranged between discs as shown in Figure 2(1), and two slits are made on the circumference of the winding barrel.

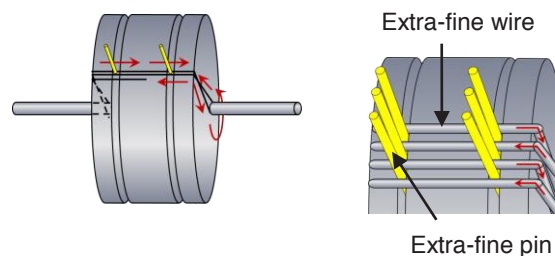
Secondly, the extra-fine wire is wound around the winding barrel in the following steps.

- 1) The end of the extra-fine wire is bound around the left side of the shaft as shown in Figure 2(2).
- 2) The extra-fine wire is tightened from the shaft to the left edge of the disc.
- 3) The extra-fine wire is bent at the left edge of the disc.
- 4) The extra-fine wire is arranged on the circumference of the winding barrel.
- 5) The extra-fine wire is bent at the right edge of the disc.
- 6) The extra-fine wire is led to under side of the right side of the shaft, and is tightened from the right edge of the disc to the shaft.
- 7) The extra-fine wire is turned around the shaft, and is tightened from the shaft to the right edge of the disc.
- 8) When the extra-fine wire is bent at the right edge of the disc again, the extra-fine wire is overlapped on the extra-fine wire which is previously tightened from the right edge of the disc to the shaft. The previously tightened wire is fixed without loosening by this overlapping and by hooking the extra-fine wire around the shaft.
- 9) The extra-fine wire is arranged again on the circumference of the winding barrel. Two extra-fine pins are inserted between previously arranged wire and just arranged wire; besides, two extra-fine wires are inserted into the slits of the circumference of the winding barrel. Arranged extra-fine wires become parallel on the circumference of the winding barrel. Gap between the arranged wires corresponds to the diameter of the extra-fine pin.
- 10) The extra-fine wire is bent again at the left edge of the disc. After that, the winding steps are similarly continued.

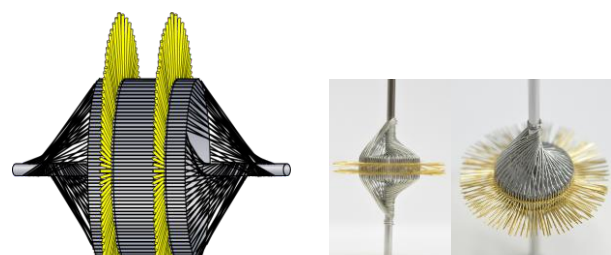
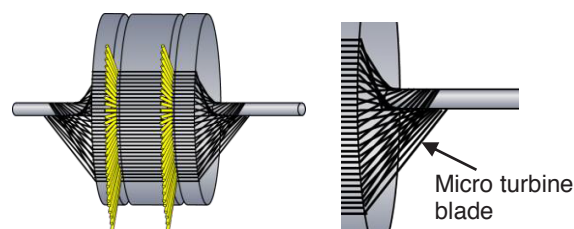
(1) The winding barrel



(2) Winding steps



(3) Winding repetition



(The end Of winding processes)

(4) Finished micro-turbine

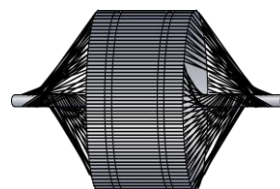


FIGURE 2. Making process of the micro-turbine impeller utilizing an extra-fine wire

Thirdly, the winding process is repeated many times. Knitted extra-fine wire forms the micro-turbine blades on both sides of the disc as shown Figure 2(3). When the arranged wires and the extra-fine pins are filled on the circumference of the winding barrel without spacing, the winding process comes to an end.

Finally, all of the extra-fine pins are removed as shown in Figure 2(4), and arranged wires forms a spur gear on circumference of the disk. The making process of the micro-turbine impeller is finished.

EXPERIMENTAL APPARATUS

Figure 3 shows the experimental apparatus for the rotation test. Compressed and cleaned air is injected to the turbine impeller from two nozzles. Flow rate of air jet is regulated by the air regulator and is measured by flow sensors; besides, the number of revolutions of the micro turbine is measured by laser sensors.

Fig. 4 shows the detail of the wind tunnel. The micro turbine impeller is located at midpoint of two conical wind tunnels. The fine shafts of both side of the micro turbine are inserted into small tube bearings, and the micro turbine is supported on the center line of two conical wind tunnels as shown in Figure 4(a). Blade angles which angles mean the angles of the extra-fine wires against the fine shaft become from 0 to 33 degrees. The nozzles are tilted 57 degrees against the fine shaft as shown Figure 4(c). Internal diameter of each nozzle is 4 millimeters; besides, the center lines of the nozzles are horizontally shifted 3 millimeters from the center line of the fine shaft as shown in Figure 4(b).

Figure 5 shows the micro turbine unit. The unit is consisted of two wind tunnels. One is a driving unit for driving the micro turbine. Another is a rotating unit used for rotating the outside micro gear. The macro turbine impeller to be integrated micro gears is also used as the outside micro gear. The small window is opened on the side face of the micro turbine unit. It is prevented by the small window that air jet flows into the rotating unit.

In experiments, the number of revolutions of the micro turbine impeller was measured without engaging the outside gears or with engaging the outside gears.

Additionally, the number of revolutions and the starting torque were measured by using different driving units in which flow angles were 57, 90, or 29 degrees.

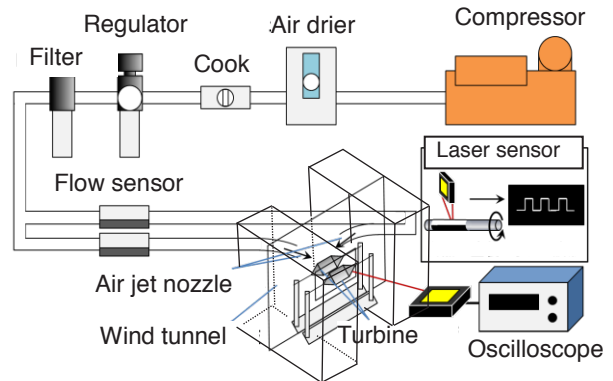


FIGURE 3. Experimental apparatus for the rotation test of the micro-turbine impeller utilizing an extra-fine wire

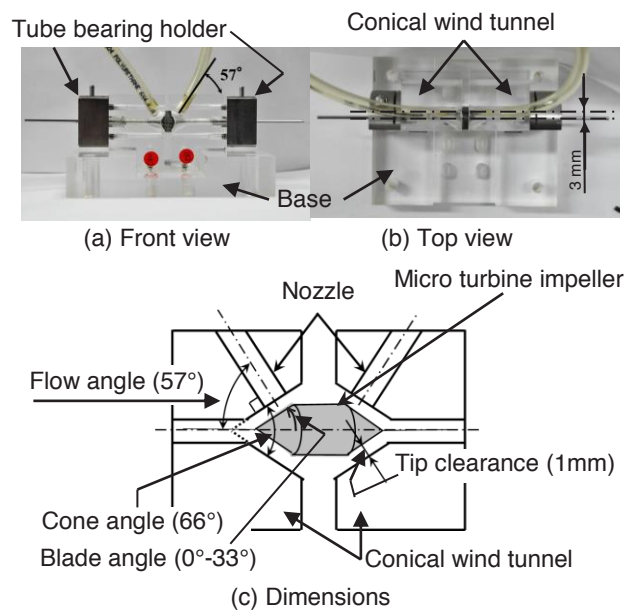


FIGURE 4. Detail of the wind tunnel

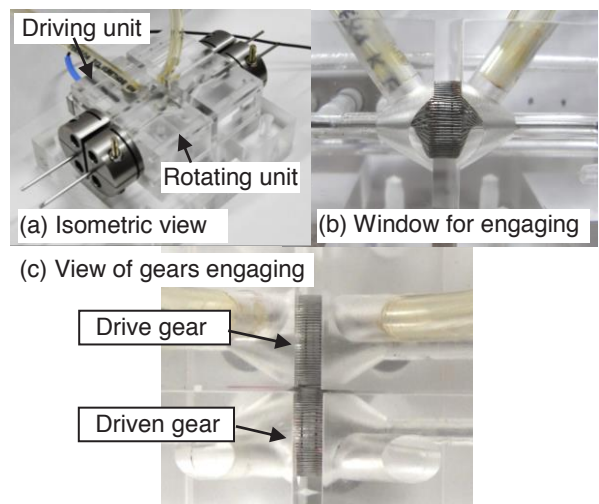


FIGURE 5. Micro turbine unit

EXPERIMENTAL RESULTS

Figure 6 showed the relation between flow rates of air jet and number of rotations of the micro turbine. The number of revolutions of the micro turbine impeller without engaging the outside gears were proportional against flow quantity. The number of revolutions of the micro turbine impeller with engaging the outside gears were lower than the number of revolutions without engaging the outside gears. Transmission ratios were calculated by dividing the revolution number of the driven gear with the revolution number of the drive gear. When the flow rate was 40 [L/min], transmission ratio nearly became 100[%]. In other flow rate, transmission ratio became the range from 85 to 98 [%]. It indicates that the micro gears were slipping.

Figure 7 showed the relation between flow rates and the number of revolutions without engaging the outside gears. When the flow angle was 57 degrees, the number of revolutions was higher than other driving unit in which flow angle was 90 or 29 degrees.

Figure 8 showed the relation between flow rates and the starting torques without engaging the outside gears. When the flow angles were 57 or 90 degrees, the starting torques were same values. These starting torques were stronger than other driving unit in which flow angle was 29 degrees. The starting torques were measured by following processes; a thread was coiled around the micro gear of the micro turbine, and small weight was hanged at the end of the thread, and the small weight was suspended from the pulley. When the flow rate was increased, the flow rates at which the small weight began to lift up were measured.

From the comparison with experiments, suitable flow angle in which air jet blew to the turbine blades was 57 degrees, for the outermost wire of the extra-fine wires crosses the air jet nozzle with 90 degrees.

CONCLUSIONS

The micro-turbine impeller integrated micro gear was developed by utilizing an extra fine. Rotational performances of the micro-turbine were confirmed by experiments.

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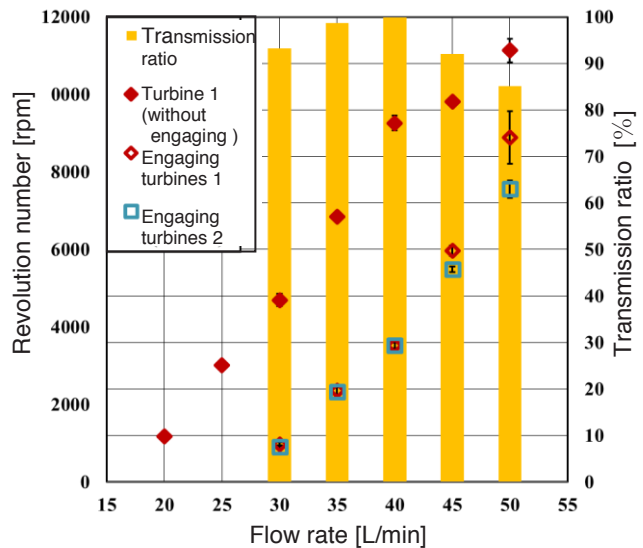


FIGURE 6. Experimental results (Relation between flow rates and number of rotations)

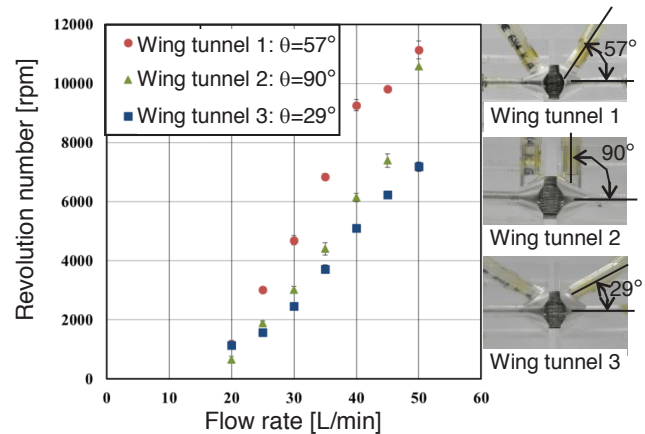


FIGURE 7. Experimental results (Relation between flow rates and number of rotations)

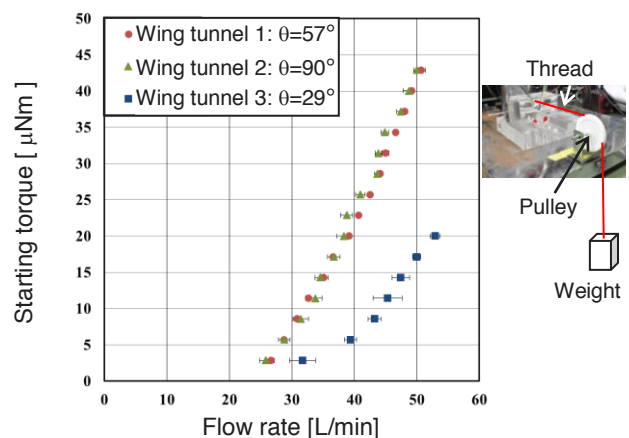


FIGURE 8. Experimental results (Relation between flow rates and starting torque)

Multi-oscillator, micro-vortex experimental apparatus

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INTRODUCTION

In this paper, the design and fabrication of an apparatus for multi-oscillator vortex studies is presented. The purpose of this is to study particle sorting and shepherding effects using vortices induced by oscillating micro-probes. Visualization and measurement of particle trajectories is achieved using microscope-based particle image velocimetry (PIV) [1]. To enable shepherding and transfer of particle clusters, multiple probes are placed into a fluid containing suspended particles. Some applications of this study includes the sorting of particles in fluids based on their geometry, size and/or density, and separating cells in biological samples based on their cell type [2].

DESIGN

The design and early investigation of micro-particle manipulation have been presented in [3, 4]. The setup represented in the previous papers can only hold one oscillating probe for single probe experiments.

In this paper a new apparatus is designed to provide flexibility in the number of implemented micro-probes and their arrangement as well as providing translation during the experiments. Figure 1 shows a schematic representation of the apparatus for this development. The mounting block which bolt to the particle manipulation apparatus, holds the stepper motors and micro-probes. The lead screw via a holding mechanism (holding blocks, see figure 2) holds the micro-probes. One probe is stationary while the other two probes can be translated along two

perpendicular axes, X and Y . These motions are provided by two stepper motor and lead screw mechanisms, one for each axis. The moving probe's motions provide the position adjustment of the probes relative to each other. Similarly, this can provide the capability of moving the probes during the experiments, for instance two oscillating probes moving with different speeds inside the fluid. The holding blocks for the fibers are designed in such a way that provides maximum flexibility in probe orientation as well as the number of probes that they can support. As it is shown in figure 2, the mounting block have four different orientation that probes can be attached to. The stationary probe is fixed to the apparatus so it has a fixed position and direction of oscillation. However, the

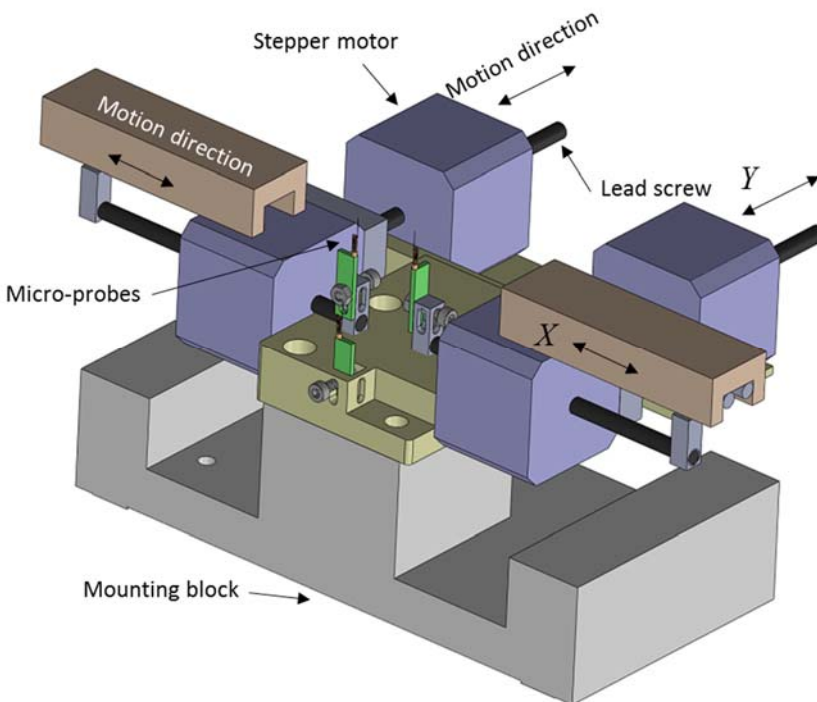


FIGURE 1. Solid model of multi-oscillator apparatus. Four stepper motors provide motion in X and Y for two moving probes.

stationary probe can also have four different orientations, based on the way it is attached to the apparatus, providing two different directions of oscillation. Moving probes can be attached to the holding blocks in a way that they oscillate either coplanar with the stationary probe or in a plane orthogonal to it with all probe axes being vertical. The moving probes that oscillate coplanar to the stationary probe's oscillation, are called in-plane moving probes. Similarly, probes oscillating perpendicular to the stationary probe's oscillation, are called out-of-plane moving probe. Besides providing 4 probe orientation options, each holding block can hold up to 4 micro-probes at the same time.

Figure 2 shows a close up view of the apparatus around the probes. In the photograph inset on the right hand side of this figure, these probes are made of a 7 μm diameter carbon fiber glued to the upper tine of a quartz tuning fork. A more comprehensive study about the dynamic response of these resonant-based micro-probes are presented in [5]. These probes are driven at their resonant frequency using a Saji waveform generator (model STQ2) to transfer the maximum energy into the fluid to create vortex flows. The left hand side probe, is a moving probe in which its oscillating plane is the same as the stationary

probe shown in the middle. The probe on the right hand side oscillates in the direction perpendicular to the other two probes, therefore it's a moving out-of-plane probe.

Figure 3 is a schematic diagram of the probes in a fluid represented by the circular reservoir. In this configuration one in-plane and one out-of-plane moving probe are used. The probes are inside the fluid with container boundary shown by the dashed line. The moving in-plane and out-of-plane probes can scan within the container having (X_1, Y_1) and (X_2, Y_2) coordinates respectively.

EXPERIMENTAL SETUP

Figure 4 shows the experimental apparatus of the designed system. Because the probes have to enter the surface of the fluid vertically from above, the mounting block of the multi-oscillator apparatus is bolted to the underside of the structural frame (figure 4-a).

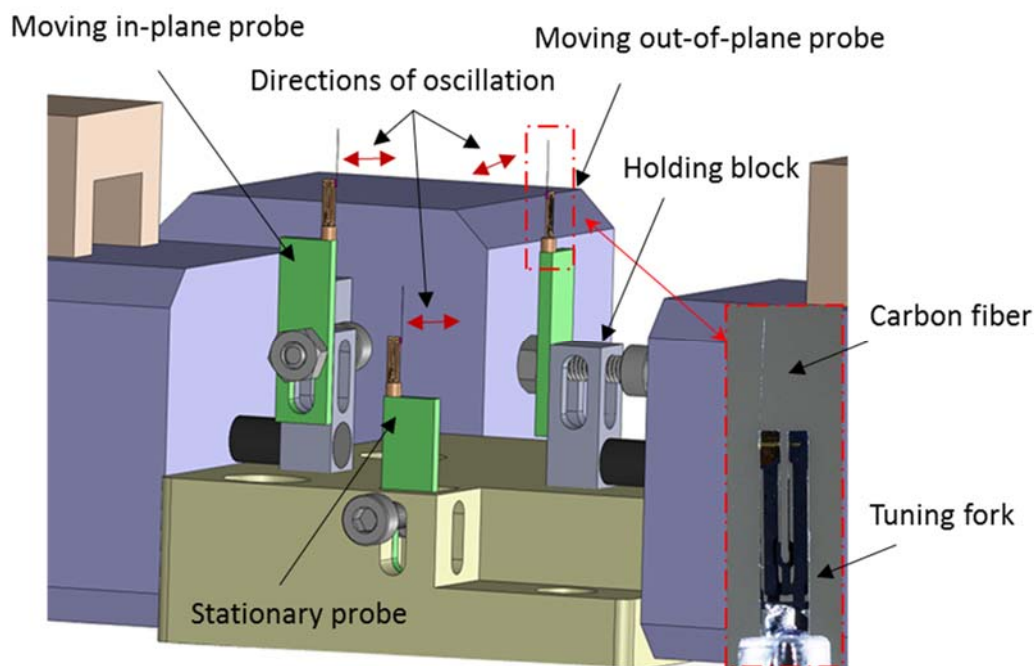


FIGURE 2. Close up view of the solid model for the micro-vortex multi-oscillator probing arrangement. Bottom right hand image illustrate the image of the micro probes containing a 7 μm diameter carbon fiber glued to a quartz tuning fork tine.

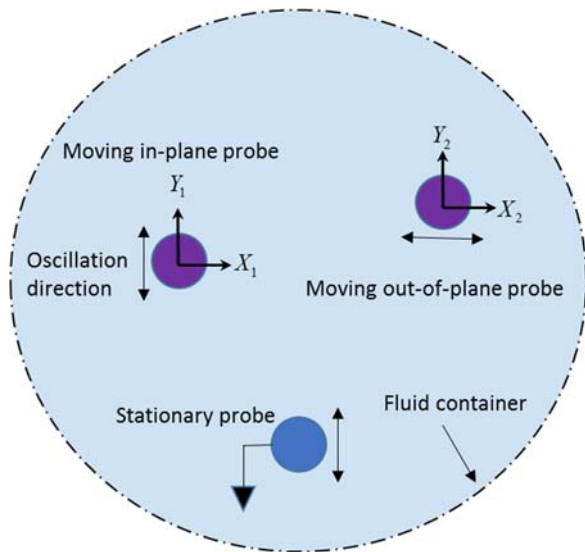


FIGURE 3. Schematic diagram of the probing system in the fluid container. The two upper moving probes are able to move anywhere inside the container along X and Y axes.

The fluid height adjustment stage is used to control the depth of the probes into the fluid container. Typically this height is set by contacting the probes with the bottom surface of the container and then retracting them 10 – 100 μm so that they can freely oscillate. The camera images the probe and particles from the underside of this container. Figure 4 shows a photograph of the micro-probes system attached to the underside of the structural frame. The two stepper motors shown in Figure 4-a) are used to position the probes before and during the

experiment to create different layouts (i.e. triangular or circular shapes). Figure 4-b) shows two probes, the lighting source and prism. One face of the prism deflects the bright light source vertically into the fluid container preventing image saturation and the other face provides a viewing mirror that reflects light scatter from the moving probes and particles back into the camera. In the photograph shown the fluid container is a cylindrical reservoir bonded to a glass slide. Depending on the probe arrangement and/or boundary condition(s) different size/types of reservoir are used.

FUTURE WORK

Theoretical modeling of the micro-particle flow patterns and comparison of these to experimental studies [6] has been initiated over the last three. A more systematic study to assess the influence of boundary conditions, particle geometry and density with multiple micro-fiber oscillators are planned. Furthermore, because of the large number of particles and noise in the camera images, the concept of novel encoded particle image velocimetry (ePIV) for extracting particle trajectories from these complex flow patterns is being explored and some preliminary studies published [7].

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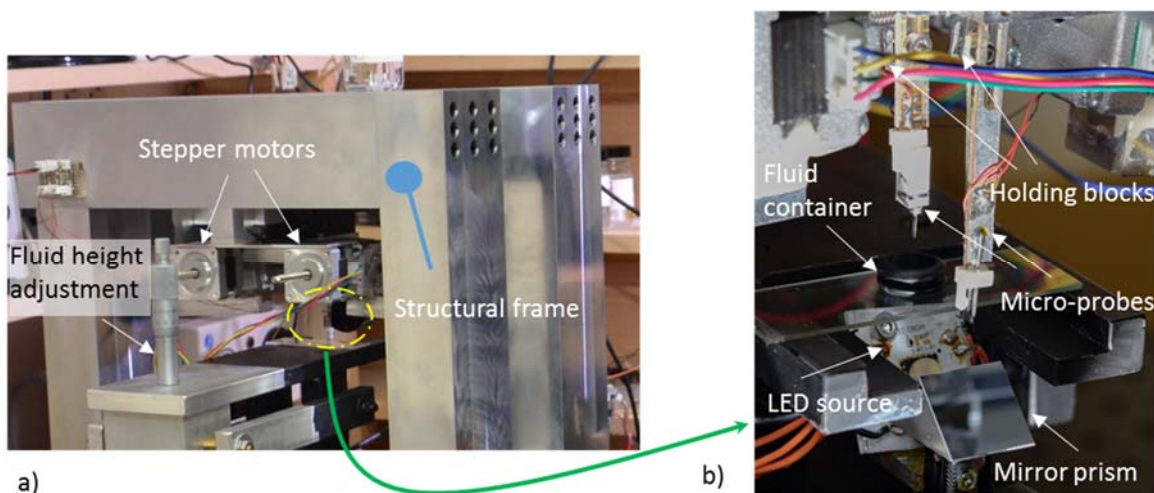


FIGURE 4. Experimental multi-oscillator apparatus. a) multi-oscillator apparatus attached to the structural frame of the particle manipulation framework [4,5]. b) Close shot of the micro-probes system.

are those of the author(s) and do not necessarily reflect the views of the National Science Foundation.

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Prediction of Motion Errors in Linear Stages under Loads

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INSTRUCTIONS

We have developed a simulation technology for 6-DOF motion errors of precision linear motion stages under unloading conditions in previous research [1]. The 5-DOF motion errors such as vertical and horizontal straightness errors, pitch, yaw, and roll were estimated from the profile errors of the guide rail using the transfer function method [2]. In case of positioning error, it was predicted from the Abbe offset principle and a scale error itself when a linear scale was used as a feedback system.

In this paper, we have improved the simulation technology considering the external loads such as workpiece weight, cutting force, and cogging force of a cored linear motor in predicting the motion errors. In order to consider the external loads, the modified transfer function method was proposed to reflect clearance changed according to the loads in a hydrostatic and aerostatic bearing table. Then, reaction forces using the modified transfer function were calculated, and the equilibrium equations were derived from the reaction forces and external loads for predicting the 5-DOF motion errors. The motion errors were predicted for hydrostatic and aerostatic, and linear motion bearing stages, and compared with the actually measured motion errors.

CALCULATION OF MOTION ERRORS

Transfer function $G(\omega)$ is a ratio of the magnitude of the reaction force $f(\omega)$ to the amplitude of the rail form error $e(\omega)$ at each spatial frequency ω [3].

$$G(\omega) = \frac{f(\omega)}{e(\omega)}$$

When loads are applied to oppositely supported hydrostatic and aerostatic bearings, equilibrium clearances are changed in the upper and lower pads, as shown in FIGURE 1. These clearances can be calculated by changing them repetitively until applied loads become total reaction forces

generated in the upper and lower pads. After the clearances are determined, the transfer functions are calculated at each clearance in the upper and lower pads, respectively. Finally, the motion errors are calculated using the modified transfer function.

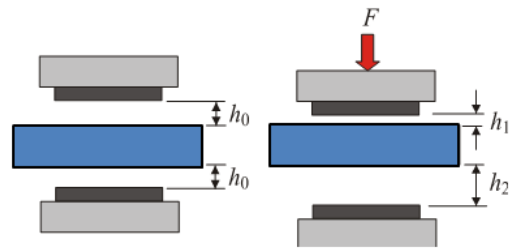


FIGURE 1. Clearance change in oppositely supported bearing after load is applied

FIGURE 2 describes the procedure for calculating motion errors under loads. In case of single pad bearing or linear motion bearing(LMB), the transfer function should be also calculated at equilibrium clearance caused by the loads.

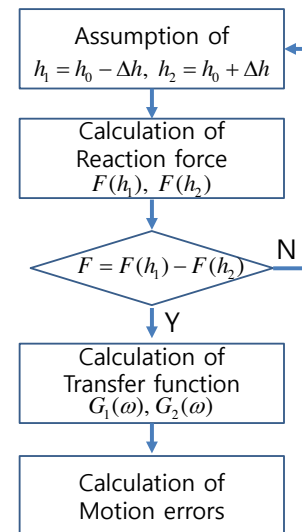


FIGURE 2. Procedure for calculating motion errors under loads

If the additional force caused by misalignment of a ballscrew or cogging force of a cored linear motor is applied to the linear motion stages, it should be also considered in the equilibrium equations of force and moment. The remaining process is the same as unloading conditions except for using the modified transfer function. FIGURE 3 shows the transfer function calculated in the upper and lower pads for a hydrostatic bearing pad with external vertical load up to 1000 kg, as an example. In the upper pad, the clearance decreased with higher load so that the reaction force certainly increased with the load. However, the lower pad has an opposite effect with respect to the load because clearance increased with the load. The reaction force decreased with higher load in the lower pad.

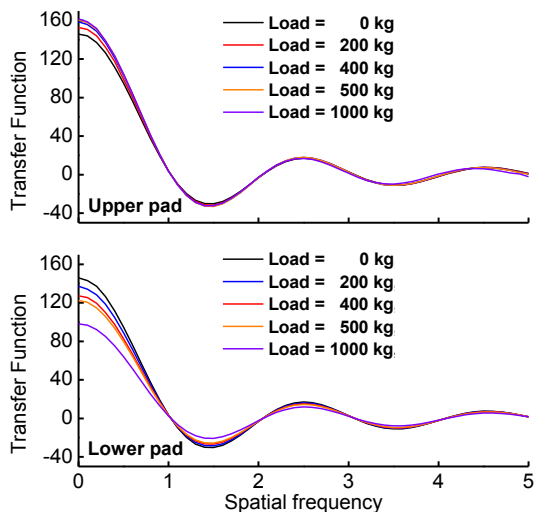


FIGURE 3. Transfer function in the upper and lower pads under loads

EXPERIMENT RESULTS

Predictions and experiments for motion errors with external loads were carried out for hydrostatic and aerostatic bearing, and linear motion bearing stages.

FIGURE 4 shows an experimental setup for a linear motion bearing (LMB) stage, as an example. The LMB stage is composed of four bearings and two rails. It is actuated using a linear motor with UMAC controller and linear scale feedback system.

In order to predict the motion errors, the rail profile errors should be firstly known. But it is difficult to measure the rail profile errors because the rail form is not flat but groove in case of linear motion bearing. Therefore we

have modified the algorithm which is to use straightness errors of one bearing block instead of rail profile errors [4]. The straightness errors of the one bearing block can be easily measured with a laser interferometer or dial gauge and straight edge.

In this experiment, workpiece weight is regarded as external load, so it is applied to the table in a vertical direction. The loads are changed to 0, 40, 80 kg. Vertical straightness and pitch error are measured with a laser interferometer.



FIGURE 4. Experimental setup

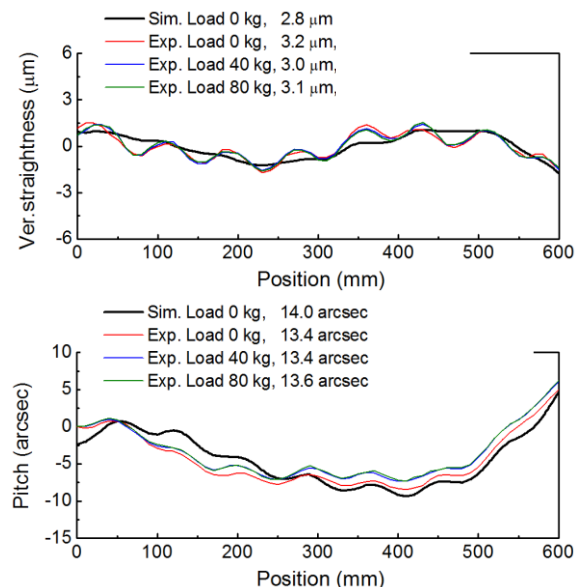


FIGURE 5. Comparison between simulated and measured motion errors with loads

FIGURE 5 shows a comparison result between predicted and measured motion errors with loads. From simulation, the motion errors are almost same regardless of the load. So simulation result for an unloading condition was

shown in the figure. In the experiments, the influence of the load on the motion errors is also negligible. The experiment results are well agreed with those of the simulation except for sinusoidal wave component. It is likely that the sinusoidal waves in experiment come from deformation of rail generated when it is fastened on the base using bolts.

Influence of additional force caused by misalignment of a ballscrew is carried out for a hydrostatic bearing stage. Adjustment device is designed in order to apply the misalignment toward a horizontal direction at supporting bearing of the ballscrew. The additional force is theoretically calculated according to the deflection of the ballscrew, and it is considered in the equilibrium equations of force and moment. Horizontal straightness error and yaw error are predicted using the measured rail profile errors and the transfer function method. The results show that they are varied with a magnitude of the misalignment of the ballscrew, and the simulated motion errors have a very similar tendency to the experimental results.

CONCLUSIONS

This paper describes a prediction of motion errors considering loads in precision linear motion stage. The loads can be workpiece weight, cutting force, cogging force caused by cored linear motor, and additional force caused by misalignment of ballscrew. The motion errors were calculated using the modified transfer function method and equilibrium equations of force and moment considering loads. The influence of loads on the motion errors were theoretically and experimentally addressed in hydrostatic and aerostatic, and linear motion bearing stages.

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6-DOF Active Vibration Isolation System with Electromagnetic Actuator using Halbach Magnet Array

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INSTRUCTIONS

There are various tiny things to be measured and handled in nanotechnology. Several equipment such as SEM, AFM, and so on, are used effectively to deal with nm-size specimens. The performance of the equipment is deteriorated due to disturbances such as temperature variation, vibration, and so on. A passive or an active vibration isolator are applied to isolate the equipment from the vibration. Especially, this paper presents an active vibration isolation (AVI). [1]

the AVI is composed of four metal spiral springs as a passive system, eight voice coil motors, eight velocity sensors for 6 degree of freedom (DOF) vibration isolation. In this paper, a voice coil motor system with a halbach magnet array and a set of coil is applied to the actuating system to make the AVI to be compact. Because of its halbach magnet array makes a higher force than the general magnet array in the same volume. In this paper, the velocity sensor, GS-11D, not to have a unity gain in low frequency is applied to measure vibrations. In this paper, a compensator is adopted to make the magnitude of the velocity sensor a unity gain. There are variations of the disturbances and the load changes of the systems and machines. Due to the variations and changes, the performance of the isolation system might be not robust. As a controller, a robust controller called H-infinity is presented. In addition, modal decomposition technique is applied. The AVI system to which the compensator and the controller are applied is implemented with a dSPACE.

6-DOF ACTIVE VIBRATION ISOLATION SYSTEM

A. Configuration of The AVI System

The configuration of the AVI system is shown in Figure 1. The AVI system places the four metal spiral springs between lower plate and upper plate to isolate from base vibration. There are eight actuators symmetrically to be located for the

simple modeling. Each four actuators are in charge of the horizontal and vertical movements. Vertical actuators take charge of vertical, rolling, pitching motion. Horizontal actuators take care of translation and yawing movements. In addition, the six velocity sensors are mounted in a symmetrical structure in the upper. And, the rest sensors attached to lower plate and upper plate for a feedforward and for its transmissibility.

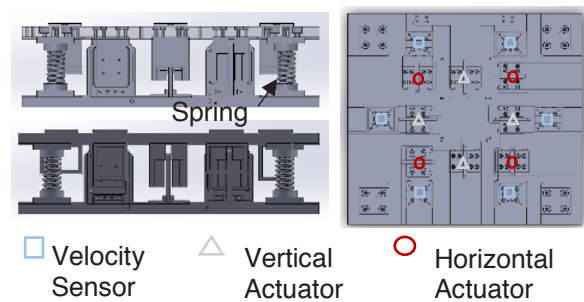


FIGURE 1. Configuration of The AVI System

Also, each actuator operates on the basis of single input-single output (SISO) control loops, which do not explicitly account for any coupling between the AVI system. This being so, in order to design the AVI system as a passive isolator with four metal springs, CAE is used and modes of the AVI system are obtained as shown in Figure 2.

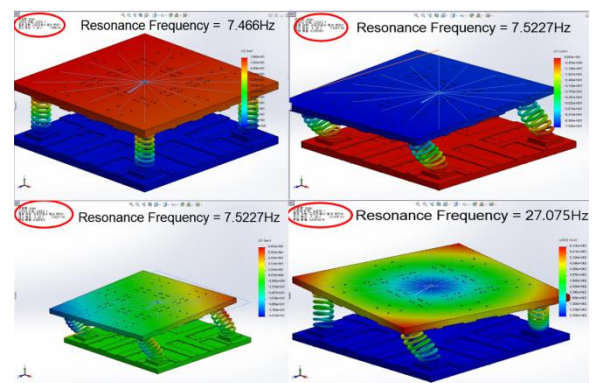


FIGURE 2. Dynamic analysis of the AVI system

The target performance of the AVI system is : payload 50kg, the transmissibility decreased 20dB at the 10Hz.

B. System Dynamics Model

The passive isolator can be modeled as follows Figure 3, using the principal axes (X, Y, Z) stiffness and damping.

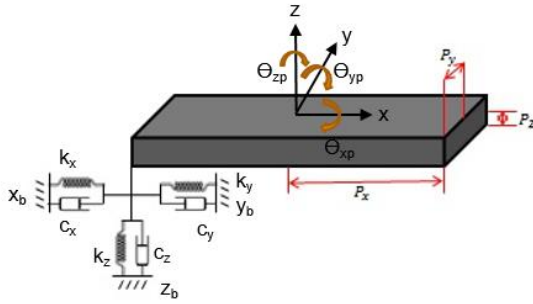


FIGURE 3. Modeling of the passive system

$$F_x = M_p \ddot{x}_p + C_x (\dot{x}_p - \dot{x}_b) + k_x (x_p - x_b) + C_x p_z (\dot{\theta}_{yp} - \dot{\theta}_{yb}) + k_x p_z (\theta_{yp} - \theta_{yb}) \quad (1)$$

$$F_y = M_p \ddot{y}_p + C_y (\dot{y}_p - \dot{y}_b) + k_y (y_p - y_b) + C_y p_z (\dot{\theta}_{xp} - \dot{\theta}_{xb}) + k_y p_z (\theta_{xp} - \theta_{xb}) \quad (2)$$

$$M_{\theta_z} = I_p \ddot{\theta}_{zp} + (C_x p_y^2 + C_y p_x^2) (\dot{\theta}_{zp} - \dot{\theta}_{zb}) + (k_x p_y^2 + k_y p_x^2) (\theta_{zp} - \theta_{zb}) \quad (3)$$

$$F_z = M_p \ddot{z}_p + C_z (\dot{z}_p - \dot{z}_b) + k_z (z_p - z_b) \quad (4)$$

$$M_{\theta_x} = I_{xx} \ddot{\theta}_{xp} - C_y p_z (\dot{y}_p - \dot{y}_b) - k_y p_z (y_p - y_b) + (C_y p_z^2 + C_z p_y^2) (\dot{\theta}_{xp} - \dot{\theta}_{xb}) + (k_y p_z^2 + k_z p_y^2) (\theta_{xp} - \theta_{xb}) \quad (5)$$

$$M_{\theta_y} = I_{yy} \ddot{\theta}_{yp} - C_x p_z (\dot{x}_p - \dot{x}_b) - k_x p_z (x_p - x_b) + (C_x p_z^2 + C_z p_x^2) (\dot{\theta}_{yp} - \dot{\theta}_{yb}) + (k_x p_z^2 + k_z p_x^2) (\theta_{yp} - \theta_{yb}) \quad (6)$$

Equations of motion for the four passive system can be described as shown in Eq. (1) to (6). To consider the applied force vector $\{F_{ud}\}$ in all these 6-DOF, the equation of motion in matrix form can be written as :

$$([M]s^2 + [C]s + [K])\{p(s)\} = ([C]s + [K])\{b(s)\} + F_{ud} \quad (7)$$

Here $\{F_{ud}\}$ is the force vector comprising of the applied control and direct disturbance

forces and moments in the respective direction.

C. Modal Decoupled Modes

Especially, in the AVI system, two motions, Z and θ_z , are decoupled and other motions are coupled. For each actuator to be operated on the SISO control loop, the model of the AVI system is decoupled by using modal decomposition technique. Both the coupled and decoupled open-loop transmissibility for its normalized six rigid body modes are shown in Figure 4.

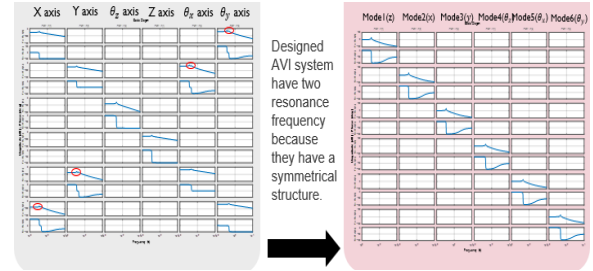


FIGURE 4. Open loop transmissibility of modal decoupled modes

CONTROLLER DESIGN

A. Analysis of System Uncertainties

In this case of 6-DOF AVI system, payload mass, damping coefficient, and stiffness of the passive isolators can be regarded as the main uncertainties. As using a AVI system, a mass of a measuring equipment to be placed on the AVI system is not constant, so payload mass can be much more dominant than the other parameters.

Based on practical considerations, the mass and moments of inertia $\{m, I_{xx}, I_{yy}, I_{zz}\}$ have relative errors about 20% around the nominal values.

Therefore, the actual mass and moments of inertia are presented as

$$m = \bar{m}(1 + u_m \delta_m) \quad (8)$$

$$I_{xx} = \bar{I}_{xx}(1 + u_m \delta_m) \quad (9)$$

$$I_{yy} = \bar{I}_{yy}(1 + u_m \delta_m) \quad (10)$$

$$I_{zz} = \bar{I}_{zz}(1 + u_m \delta_m) \quad (11)$$

Here $\bar{m}$, $\bar{I}_{xx}$, $\bar{I}_{yy}$, and $\bar{I}_{zz}$ are each nominal mass and nominal moments of inertia. u_m has a value of 0.2 on the mass relative uncertainty and $-1 \leq \delta_m \leq 1$. In addition, damping and stiffness also can be modeled in a similar way.

$$c_x = c_y = c_{xy} = \bar{c}_{xy}(1 + u_c \delta_c) \quad (12)$$

$$c_z = \bar{c}_z(1 + u_c \delta_c) \quad (13)$$

$$k_x = k_y = k_{xy} = \overline{k_{xy}}(1 + u_k \delta_k) \quad (14)$$

$$k_z = \overline{k_z}(1 + u_k \delta_k) \quad (15)$$

The linear fractional transformations are extracted for the $\delta_m, \delta_c, \delta_k$ as follows.

$$Q_m = \begin{bmatrix} -\overline{M_p}^{-1} M_{p_u} & \overline{M_p}^{-1} \\ -\overline{M_p}^{-1} M_{p_u} & \overline{M_p}^{-1} \end{bmatrix} \quad (16)$$

$$Q_c = \begin{bmatrix} 0_{6 \times 6} & I_{6 \times 6} \\ C_{p_u} & \overline{C_p} \end{bmatrix} \quad (17)$$

$$Q_k = \begin{bmatrix} 0_{6 \times 6} & I_{6 \times 6} \\ K_{p_u} & \overline{K_p} \end{bmatrix} \quad (18)$$

$$\begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \\ \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \\ y_m \\ y_c \\ y_k \\ y \end{bmatrix} = \begin{bmatrix} A_1 & B_1 & B_2 \\ C_1 & D_{21} & D_{22} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \\ \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \\ u_m \\ u_c \\ u_k \\ u \end{bmatrix} \quad (19)$$

System block diagram is shown in Figure 5.

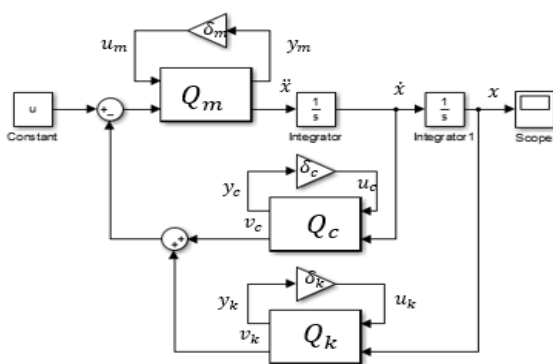


FIGURE 5. Block diagram of the passive isolation system with uncertainty parameter

In the Figure 5, the input-output relationship in the periphery of each parameter perturbation, get the governing equations of the system are shown in Equation (19). Also, $\|\Delta_\infty\| \leq 1$ because, it is $-1 \leq \delta_m, \delta_c, \delta_k \leq 1$.

B. Design of H-infinite Controller

The block diagram of the earlier obtained based on the results of the modeling can be indicated total uncertainty closed-loop system.

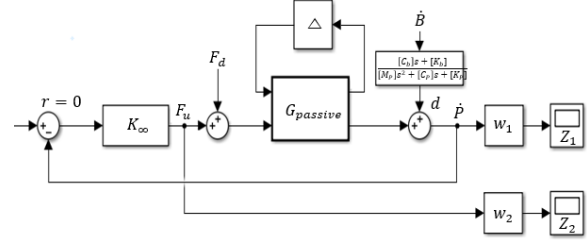


FIGURE 6. Closed-loop system structure

Where $\dot{P}$ represents 6-DOF velocities of the payload to the output signal measuring in the experiment. And $\dot{B}$ represents 6-DOF velocities of ground vibration, F_d is a direct disturbance force transmitted directly to the payload. The purpose of this controller is to ensure at least the effect of $\dot{B}$ on $\dot{P}$. In the case of Z_1, Z_2 is the respectively $\dot{P}$ of the measured output velocity multiplied the weighting function W_1, W_2 . And the disturbance can be defined as d and obtaining a relationship between Z_1, Z_2 and d are as follows.

$$\begin{bmatrix} Z_1 \\ Z_2 \end{bmatrix} = \begin{bmatrix} W_1(I + GK_\infty)^{-1} \\ W_2K_\infty(I + GK_\infty)^{-1} \end{bmatrix} \quad (20)$$

S is generally referred to as sensitivity function. After all, a matter of design H-infinity controller is defined as follows. [2]

$$\left\| \begin{bmatrix} W_1 S \\ W_2 K_\infty S \end{bmatrix} \right\|_\infty < \gamma \quad (21)$$

In other words, the sensitivity function between the disturbance and the measured output is determined from W_1 to have the desired damping form at the low frequency band. In the case of W_2 as role in prevent saturation of the actuator to the weighting function of the control input F_u . It is mainly defined by a constant. [3]

SIMULATION EXPERIMENT

A. Sensor Compensate Filter

The velocity meter, GS-11D, has used to measure vibrations in the AVI system. But it has not a unity gain in low frequency, below 4Hz. The transfer function of the GS-11D can be calculated through the relationship between the output voltage and the input velocity value. The transfer function has been obtained for GS-11D as shown in Figure 7. In order to compensate for this characteristic, it was applied to design a compensate filter with three filters as shown in Figure 7.

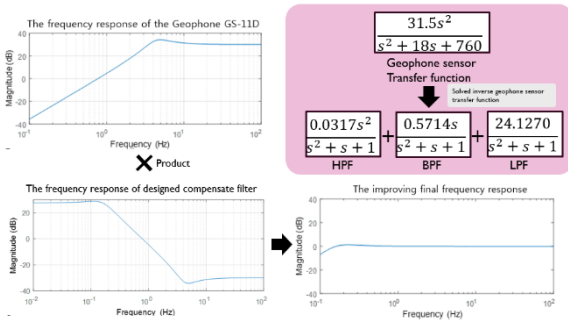


FIGURE 7. Sensor compensate filter

B. Simulation

H-infinity controller was obtained by using MATLAB, where each element is represented by the transfer function in the 6x6 matrix form. The simulation is applied to the controller designed as a SISO system in each mode. And results of simulation transmissibility of the AVI system including perturbations value are verified as shown in Figure 8.

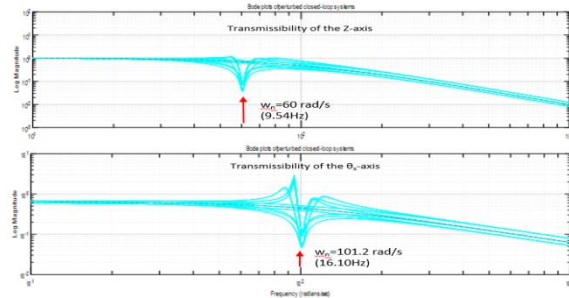


FIGURE 8. Transmissibility of applied each perturbation values

C. Experiment

In order to operate the AVI system, dSPACE (DS1007) calculates the signal received from velocity meter. Accordingly, the output signal at the dSPACE via the amplifier(TA-115) amplifies and drives the VCMs as shown in Figure 9.

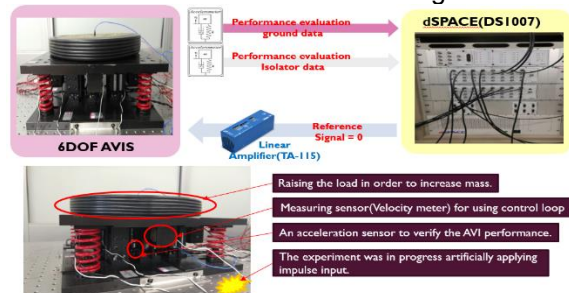


FIGURE 9. System setup for experiment of the AVI system

As shown in Figure 10, the experimental results show that the proposed AVI system can significantly reduce transmissibility from 0dB to 35dB at its resonance frequency.

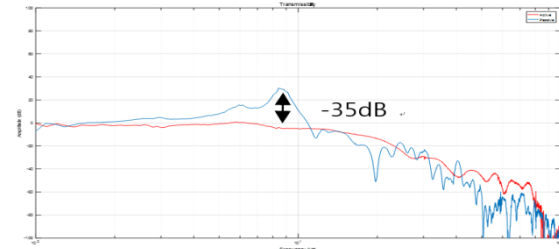


FIGURE 10. Transmissibility of AVI system

CONCLUSIONS

This paper proposes a 6-DOF AVI system with voice coil motors using halbach magnet array. The controller applied to each mode the H-infinity considering the system uncertainties, aligning the requirements applicable to the experiment, and taking account of the influence of the noise.

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Error Modeling for Calibration of the Tri-Pyramid Parallel Robot

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INSTRUCTIONS

In this paper, a simplified error model was applied to the three dof (degree of freedom) translational parallel robot, the Tri-pyramid robot. First, the total kinematic errors were classified as joint errors, kinematic parameter errors, and actuator control errors. The number of the total error is 90, and this full error model is made by using linearized Homogeneous Transformation Matrix (HTM) and DH parameter. In addition, the 27 dominant error sources affecting the accuracy of the platform position were selected through a sensitivity analysis method. Then, the calibration process is progressed and the accuracy can be improved by 70% on the basis of the reduced error model compared to the nominal kinematic model which has no error by simulation.

The Tri-pyramid robot

The tri-pyramid parallel robot was developed for an incremental forming process. For this process, high stiffness, accuracy, and X, Y, and Z translational motions are required to resist the reaction forces. A parallel mechanism was adopted to ensure stiffness. Each of the three subchains comprised four one-DOF joints without universal or spherical joints to enhance both stiffness and accuracy. Three revolute joints were connected to the tool platform so that its size was kept compact. Three-dimensional (3-D) translational motion was achieved without rotation using two parallel rotational axes.

The tri-pyramid parallel robot, shown in Figure 1, comprised three identical serial chains, symmetrically arranged on the base frame at 120° intervals. The end of each chain was connected to the robot's tool platform at 120° intervals. Each serial chain was formed from two prismatic, and two revolute, joints. The tool platform was driven by three linear actuators attached to the base frame. The tool platform could translate in the X Y, and Z directions, and had no rotational motion. Because rotational

motion does not occur, the rotational axes were parallel to each other and the two rotation angles were equal ($\theta_{i,u} = -\theta_{i,d}$) [1].

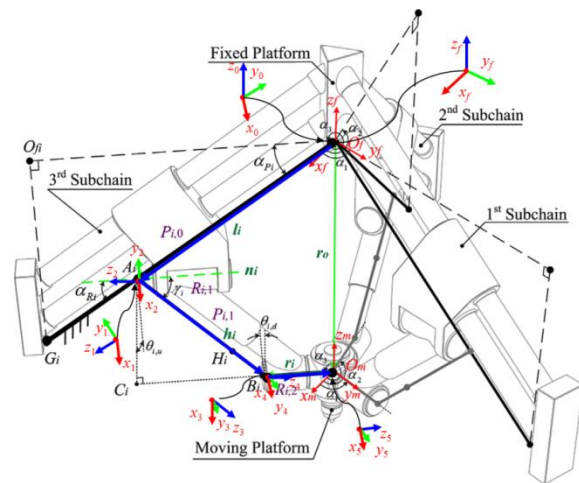


Figure 1. Schematic diagram of the forward kinematics of the tri-pyramid parallel robot.

Kinematic equations of the tri-pyramid robot

A fixed reference frame $O_f(x_f, y_f, z_f)$ is located at the origin of the robot, and a moving reference frame $O_m(x_m, y_m, z_m)$ is at the center of the moving platform. G_i ($i = 1, 2, 3$) is the end point of the linear actuator, l_i and O_{fi} is the projected point on the plane $x_f y_f$ from G_i along the axis z_f . The angles between the vectors, $O_f O_{fi}$ and axis x_f about axis z_f are α_1 , α_2 and α_3 respectively. The intersection angles between vector $O_f G_i$ and plane $x_f y_f$ are α_{Pi} , and prismatic joint $P_{i,0}$ is located on the vector $O_f G_i$. The revolute joint $R_{i,1}$, the prismatic joint $P_{i,1}$, and the revolute joint $R_{i,2}$ are placed in the order named. The angle α_{Ri} is defined as the angle between the directional vector of $P_{i,0}$ and the rotational axis of $R_{i,1}$, and γ_i is the angle between the rotational axis of $R_{i,1}$ and the directional vector of $P_{i,1}$. The length of vector r_i is the radius of the tool platform. The linear and rotational values of $P_{i,0}$, $R_{i,1}$, $P_{i,1}$, and $R_{i,2}$ are l_i , $\theta_{i,u}$, h_i , and $\theta_{i,d}$, respectively; l_i is an

active joint value, and the others are passive joint values. The vector, r_o is the position of O_m in the coordinate of reference frame O_t . The tool platform position r_o can be calculated from the forward kinematic equation.

The DH parameter rule is used to formulate the kinematic equation of each subchain [2], and the geometric parameters are listed in Table 1.

Table 1. Denevit–Hartenberg (DH) parameters of the geometric parameters [1]

i	α_{i-1}	a_{i-1}	d_i	θ_i
0	0	0	0	$\alpha_i + 90$
1	$90 - \alpha_{Pi}$	0	l_i	0
2	α_{Ri}	0	0	$\theta_{i,u}$
3	$180 + \gamma_i$	0	h_i	0
4	$-\gamma_i$	0	0	$-\theta_{i,u}$
5	0	0	r_i	0

The forward kinematic equation of the i^{th} chain in terms of its DH parameter is given by [2]:

$$P_{ti}(q_u, q_v, \rho) = {}^0T_i {}^0T_i {}^1T_i {}^2T_i {}^3T_i {}^4T_i {}^5T_i \times [0 \ 0 \ 0 \ 1]^T, \quad (i = 1, 2, 3)$$

$$q_u \equiv \{l_i\}, \quad q_v \equiv \{\theta_{i,u}, h_i, \theta_{i,d}\},$$

$$\rho \equiv \{\alpha_i, \alpha_{Pi}, \alpha_{Ri}, \gamma_i, r_i\}, \quad (1)$$

where, P_{ti} is the tool platform position and jT_i is the homogeneous transformation matrix (HTM) between the j^{th} and $j+1^{th}$ coordinate frame in the i^{th} subchain. q_u , q_v and ρ are the sets of active joint variables, passive joint variables, and the parametric variables of the robot, respectively. Thus, the forward kinematics of the parallel mechanism can be calculated by imposing a constraint on the ends of the three serial subchains. The end points of the three subchains, $P_{ti}(x_i, y_i, z_i)$ ($i = 1, 2, 3$) are the same and these are represented by equation (2):

$$g(q_u, q_v, \rho) = \begin{bmatrix} x_1 - x_2 \\ x_1 - x_3 \\ y_1 - y_2 \\ y_1 - y_3 \\ z_1 - z_2 \\ z_1 - z_3 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad (2)$$

Nine variables exist in this constraint equation because there are 12 joints, and two rotation angles in each subchain have the same value ($\theta_{i,u} = -\theta_{i,d}$) [1]. As six constraint equations [Eq. (2)], and three independent joint values q_u , [Eq. (1)], are given, it is possible to calculate the value of the nine variables. The position of the

platform is known from the joint values. This means that solving the forward kinematics with nominal values is possible when all active joint variables are known.

ERROR MODELING

The transformation matrix between coordinate frames, ${}^{i-1}T_e$ includes three kinds of error factors caused by joint clearance errors, kinematic parameter errors, and actuator control error. To determine the relationship between the successive coordinate frames, the coordinate error model is developed by the implementation of the modified D-H error model for serial multi-axis mechanisms. Based on the modified D-H rule, the error model between successive coordinates is expressed in HTM form with 10 error components including six joint error components and three error components related with kinematic parameter and one control error parameter. The transformation matrix that includes the aforementioned errors, ${}^{i-1}T_e$ can be defined in HTM form as:

$${}^{i-1}T_e = R(\hat{X}_i, \alpha_{i-1} + \Delta\alpha_{i-1}) Tr(\hat{X}_i, a_{i-1} + \Delta a_{i-1}) HTM(\varepsilon_x, \varepsilon_y, \varepsilon_z, \delta_x, \delta_y, \delta_z) R(\hat{Z}_i, \theta_i + \Delta\theta_i) Tr(\hat{Z}_i, d_i + \Delta d_i)$$

$$HTM(\delta_x, \delta_y, \delta_z, \varepsilon_x, \varepsilon_y, \varepsilon_z) = \begin{bmatrix} 1 & -\varepsilon_z & \varepsilon_y & \delta_x \\ \varepsilon_z & 1 & -\varepsilon_x & \delta_y \\ -\varepsilon_y & \varepsilon_x & 1 & \delta_z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3)$$

where HTM stands for the Homogeneous Transform Matrix, $Tr(X, A)$ is the translation matrix of distance A along the X -axis, $R(X, A)$ is the rotation matrix of angle A about the X -axis. This error modeling is applied to Tri-pyramid robot[1].

SENSITIVITY ANALYSIS

The factors that affect the accuracy of the motion are the joint clearance errors, kinematic parameter errors, and actuator control errors. Firstly, joint clearance errors that exist in each joint of each subchain are considered for the sensitivity analysis. There are three rotational errors (ε_{xij} , ε_{yij} , ε_{zij} , $i=1\sim3$, $j=1\sim4$) and three translational errors (δ_{xij} , δ_{yij} , δ_{zij}) for each joint. Secondly, the five kinematic parameters, i.e., (α_i , α_{Pi} , α_{Ri} , γ_i , r_i , $i=1, 2, 3$), are defined for each subchain structure. Finally, actuator control error

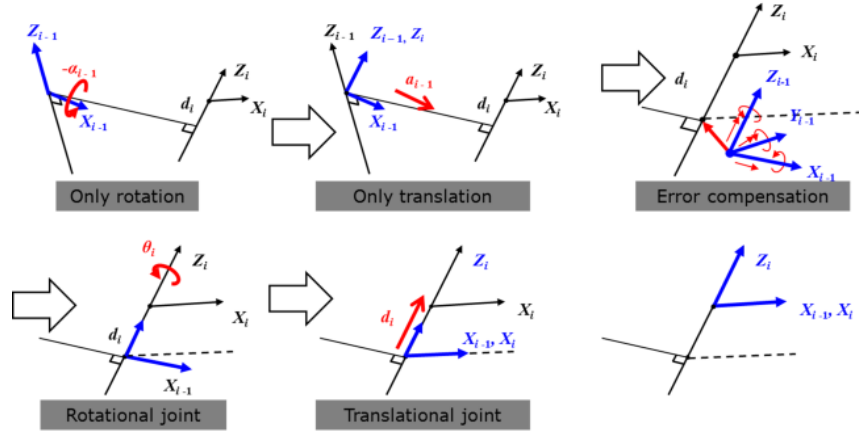


FIGURE 2. Schematic diagram of modified DH parameter

Table 2. Kinematic error components

Content	Chain $i(i=1,2,3)$
Link length	Δr_i
Angle	$\Delta \alpha_i, \Delta \alpha_{Pi}, \Delta \alpha_{Ri}, \Delta \gamma_1$
Prismatic joint error	$\varepsilon_{xi1}, \varepsilon_{yi1}, \varepsilon_{zi1}, \delta_{xi1}, \delta_{yi1},$
Rotational joint error	$\varepsilon_{xi2}, \varepsilon_{yi2}, \varepsilon_{zi2}, \delta_{xi2}, \delta_{yi2},$
Prismatic joint error	$\varepsilon_{xi3}, \varepsilon_{yi3}, \varepsilon_{zi3}, \delta_{xi3}, \delta_{yi3},$
Rotational joint error	$\varepsilon_{xi4}, \varepsilon_{yi4}, \varepsilon_{zi4}, \delta_{xi4}, \delta_{yi4},$
Actuator control error	Δl_1

is modeled for each actuator (l_i) of each subchain. Hence, there are 30 error variables for each subchain and 90 error variables for the whole Tri-pyramid parallel robot.

Assuming infinitesimal displacements/variations in each variable and comparing the normalized values of the platform displacement, it is possible to perform a sensitivity analysis. The deviation of platform position can be calculated

by end effector. Only one error variable is considered for sensitivity analysis while the remaining 89 variables are put to zero, and this set is used as an input parameter. By doing this, the influence of the other error factors on the platform accuracy can be ignored. The sensitivity coefficient can be shown in terms of normalized non-dimensional values caused by the linear motion error components:

$$\text{Sensitivity coefficient} = \frac{\|dP_t(\Delta e_l)\|}{\|\Delta e_l\|}$$

$$\Delta e_l \equiv \{\Delta l_i, \Delta r_i, \delta_{xij}, \delta_{yij}, \delta_{zij}\}, (i = 1,2,3, j = 1,2,3,4), \quad (4)$$

Totally, 27 the most sensitive parameters are selected through the sensitivity analysis. The parameters are $\Delta r_1, \Delta r_2, \Delta r_3, \Delta \alpha_1, \Delta \alpha_2, \Delta \alpha_3, \Delta \alpha_{P1}, \Delta \alpha_{P2}, \Delta \alpha_{P3}, \Delta \alpha_{R1}, \Delta \alpha_{R2}, \Delta \alpha_{R3}, \Delta \gamma_1, \Delta \gamma_2, \Delta \gamma_3, \delta_{y12}, \delta_{y22}, \delta_{y32}, \delta_{y14}, \delta_{y24}, \delta_{y34}, \delta_{y13}, \delta_{y23}, \delta_{y33}, \Delta l_1, \Delta l_2, \Delta l_3$.

CALIBRATION

In general, the forward kinematics of a parallel mechanism can be calculated using the active joint values (l_1, l_2, l_3) by numerical analysis

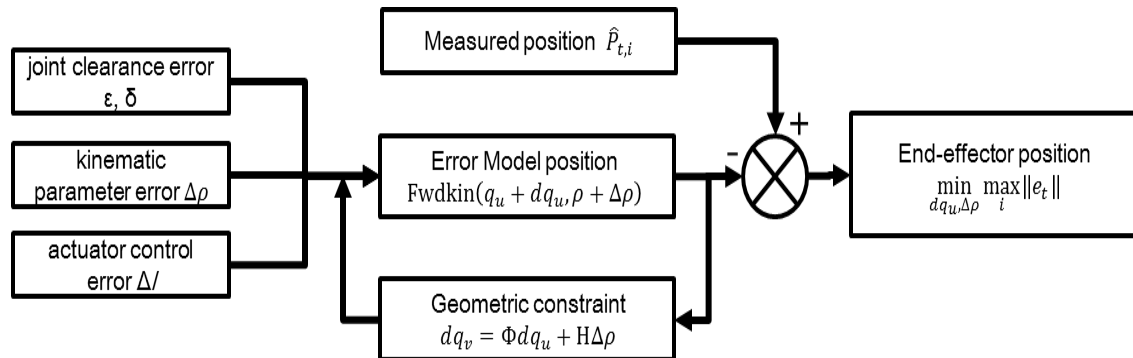


Figure 3. The calibration algorithm

methods. By using the forward kinematics, the objective function can be calculated as minimizing the difference between the measured position and the kinematic error model. Therefore, the objective function can be defined as minimizing the error value among the maximum displacement error points throughout the whole workspace. The optimization cost function can be configured as:

$$\min(e_t) = \min(\hat{P}_t - (P_t + dP_t)) \quad (5)$$

where, e_t is the end-effector error between nominal model and error model.

In addition, the 27 dominant error sources affecting the accuracy of the platform position were selected through a sensitivity analysis method in previous section. After the reduced error model based on the 27 dominant error sources are established, the calibration was carried out by simulation. The calibration simulation result says that the accuracy can be improved by 70% on the basis of the reduced error model compared to the nominal kinematic model which has no error.

CONCLUSION

As described above, the generalized error model is defined including joint clearance error, kinematic parameter error, and actuator control error. By applying an error to the error model, sensitivity analysis of each parameter can be executed to check the platform's accuracy. Based on the results of the sensitivity analysis, the dominant error parameters could be selected that affect the accuracy of the machine. Simulation result of the kinematic calibration show that it is possible to enhance the accuracy with the reduced error model by 70%.

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Design of three-jaw compliant parallel gripper for alto-alignment in vacuum environment

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INSTRUCTIONS

With the rapid progress of technology and metrology, severe working environments with vacuum, cryogenic and isothermal conditions become prerequisite nowadays. Because of operational process and space limitation in vacuum chamber, manual sample manipulation or replacement is time- and energy-consuming task. For measurement operations in vacuum chamber, an automatic sample manipulator with controllable gripper is essential for enhancing efficiency and productivity. In order to effectively maintain vacuum condition, all devices used in chamber require meticulous care to avoid construction with capillary seams and usage of easy outgassing materials. Since the late 1990s, compliant flexural manipulators with piezoelectric actuators have been developed for dedicate micro- and nano-scale operations in diverse application fields. Because of output power and size, most of commercial grippers are not developed for handling mm-scale samples. However, common lever amplifying mechanism with normal bearing also has to confront with bearing clearance and friction, which deduce asynchronous gripping force and motion.

To sequentially replace mm-scale rods from sample case to measurement holder in the restricted vacuum space, compact actuator and lubricant-free parts are preferred. To realize self-centering gripping without vision-assistant control, a concept of three-jaw compliant gripper with parallel opening motion is proposed and developed.

CONFIGURATION OF THE GRIPPER

As shown in figure 1, the developed three-jaw gripper consists of the linear actuator, flexural arms, and gripper heads. The homemade linear actuator[1] is the main driving power that can provide mm-scale movement with submicron precision. The three flexural arms are made of compact bridge-type mechanism in order to transfer the linear actuation into parallel gripping movements of three jaws. The bridge-type mechanism is realized by a compliant flexural construction due to its large motion amplification ratio, compactness, and no friction[2].

Three bridge-type flexural arms are fixed with the upper and lower blocks, as shown in figure 1(a).

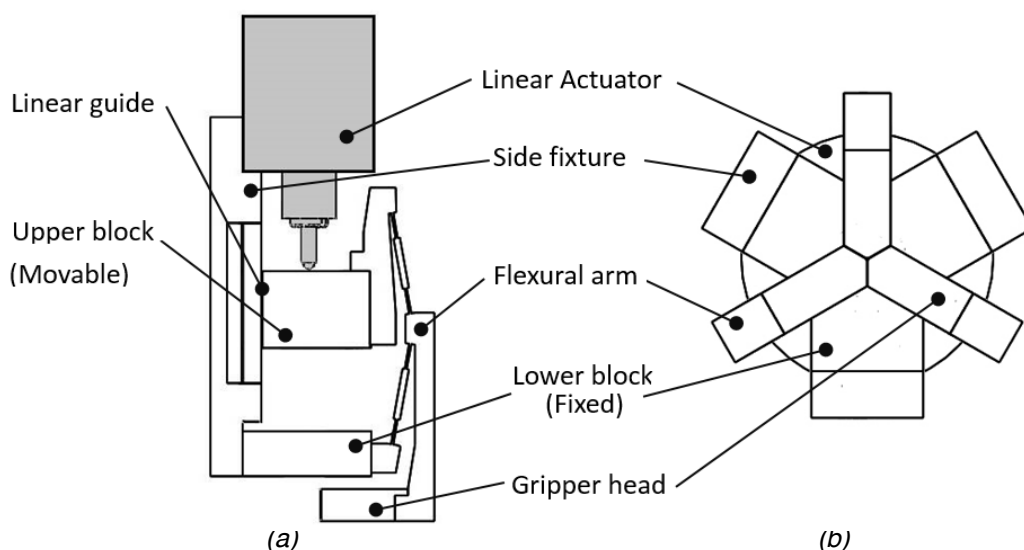


FIGURE 1. The (a)construction and (b)bottom view of the three-jaw compliant parallel gripper.

The upper block is driven by the linear actuator, while the lower block is unmovable and firmly connected with the linear actuator through three side fixtures. Furthermore, three gripper heads installed in the middle of the flexural arms are synchronically actuated to grip or release 1-3 mm sample rods.

THE KINEMATIC MODEL OF FLEXURAL ARMS

The geometry of flexure arms is the main factor affect the performance of the gripper. We applied the method of Pseudo-Rigid-Body Model (PRBM)[3] to realize effect of geometric parameter. A kinematic model of the flexural arm in figure 1(a) was developed in a simplified geometric form, as shown in figure 2. Trapezoid ABCD is the model of original arm while A'B'C'D is the state followed by an actuating displacement δy on pivot A.

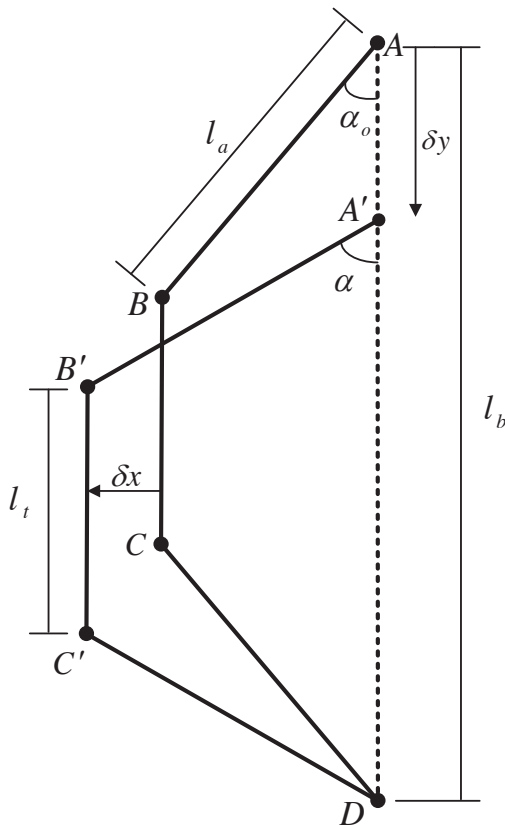


FIGURE 2. Simple geometry of flexural arms

We assume four flexural pivots A, B, C, and D have the identical spring constant k , and they would generate same angular movements and moment on pivots. We can get the relationship

between initial arm angle α_o and instant arm angle α .

$$\alpha = \cos^{-1} \left(\frac{l_b - l_t - \delta y}{l_b - l_t} \cdot \cos \alpha_o \right) \quad (1)$$

The instantaneous amplification ratio R_{amp} of actuating displacement δy to output displacement δx can be expressed as a form of α [4].

$$R_{amp} = \frac{dx}{dy} = \frac{1}{2} \cot \alpha \quad (2)$$

Combining above two equation, we can gain the relationship from δy and α to output δx .

$$\delta x = \int_0^{\delta y} \frac{1}{2} \cot \left(\cos^{-1} \left(\frac{l_b - l_t - y}{l_b - l_t} \cdot \cos \alpha_o \right) \right) dy \quad (3)$$

The linear actuator in the configuration has maximum driving force of only 100 N. Thus driving force could be a concern.

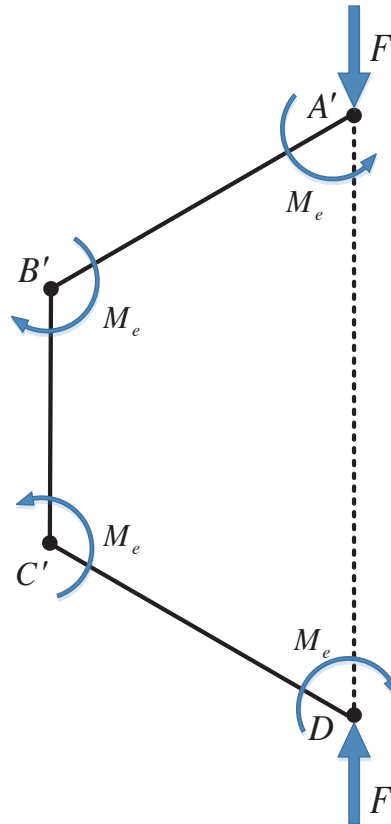


FIGURE 3. Free body diagram of flexural arms.

The relationship between driving force and moment can be derived from the law of conservation of energy. The free body diagram

can be described as shown in figure 3. Each pivot subjects to an equivalent moment M_e with different direction. Two kinds of force with identical magnitude but opposite direction F are applied on pivot A' and D.

$$\int_0^{\delta y} F dy = 4 \int_0^{\alpha - \alpha_0} M_e d\alpha \quad (4)$$

We substitute equation (3) into equation (4) and by using assumption in reference[3], an equation of driving force, output displacement and initial arm angle was obtained. E is the Young's modulus of manufactured material, I_e is the cross-sectional moment of inertia and l_e is the equivalent length of flexural pivot in equation (5).

$$F(\delta y) = \frac{4EI_e}{l_e} \left(\arccos \left(\frac{l_b - l_t - \delta y}{l_b - l_t} \cdot \cos \alpha_o \right) - \alpha_o \right) \cdot \frac{-1}{\sqrt{1 - \left(\frac{l_b - l_t - \delta y}{l_b - l_t} \cdot \cos \alpha \right)^2}} \cdot \frac{-1}{l_b - l_t} \cdot \cos \alpha \quad (5)$$

Figure 4 is the three dimensional plot of the result that substitute equation (3) into equation (5). In figure 4. we take $l_b - l_t = 30$ mm as parameter.

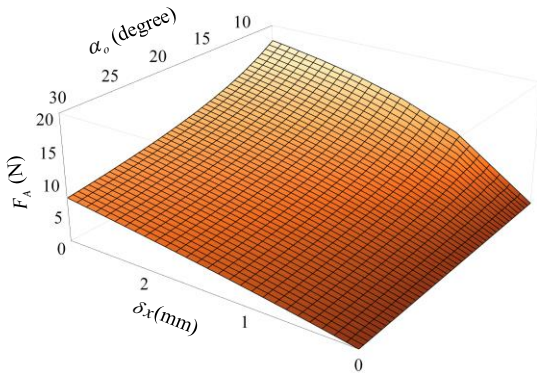


Figure 4. 3d plot of relationship between output displacement δx , initial arm angle α_o and driving force F .

FINITE ELEMENT MODEL

In the kinetic model, stress was not considered. In order to evaluate the yielding which is often the most significant failure mode in compliant mechanism, finite element models were developed by using ABAQUS/Standard finite element solver. Due to symmetry, a single flexural arm was simulated to reduce the computational cost. Eight-node quadratic shell element with reduced integration (S8R) was

chosen to mesh the mechanism. Based on mesh sensitivity studies, the total number of elements was approximately 1,300 for each model as shown in Figure 5. The material of the arm was chosen to be high grade aluminum alloy (Al 7075T6). The material properties were modeled as an isotropic, linear-elastic material with a Young's modulus of 71700 MPa, Poisson's ratio of 0.33, and yield stress S_y of 503 MPa[5]. In order to investigate the effect of initial angle to stress distribution, the initial angle was varied from 6 to 20 degree. A downward displacement δy was applied to the upper block to make the output displacement δx of 2 mm; meanwhile, the lower block was constrained in all directions.

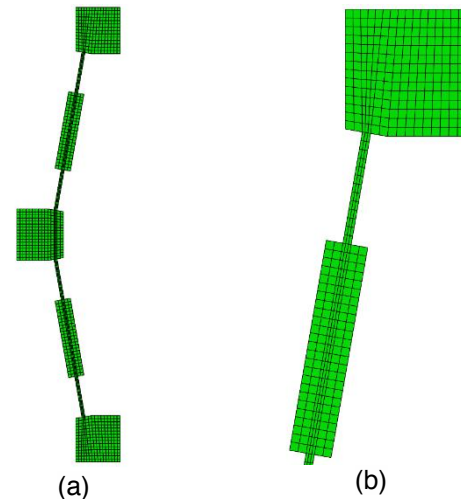


FIGURE 5. (a)Overall and (b)upper part mesh on gripper arm in ABAQUS

Figure 6 suggests that initial angle variation would not strongly affect the maximum stress σ_{max} in the flexural arm.

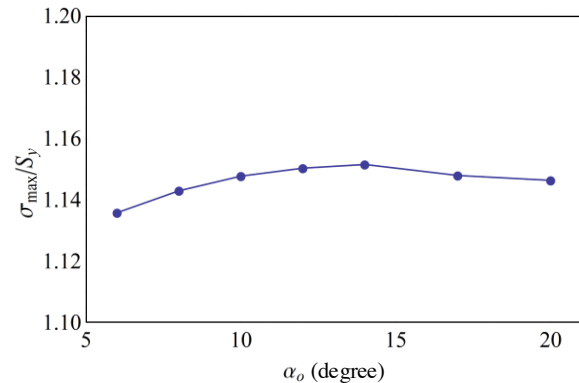


FIGURE 6. Relationship between nondimensionalized stress and initial angle with output displacement $\delta x = 2$ mm

PROTOTYPING

The flexural arms made of high grade aluminum alloy are precisely machined by electric discharge machining (EDM). The gripper heads were made of polyether ether ketone (PEEK) to have softly haptic gripping. Figure 7 demonstrates the developed prototype of three-jaw compliant parallel gripper.

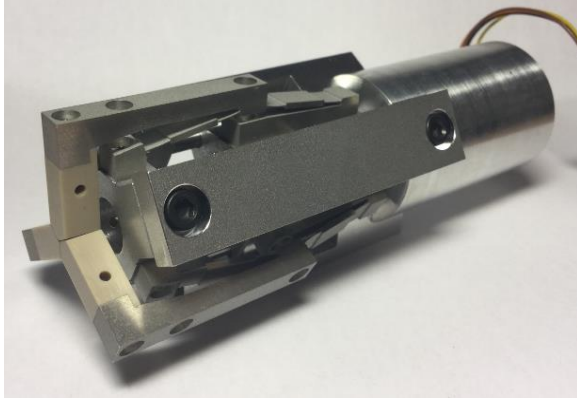


FIGURE 7. Developed prototype of three-jaw compliant parallel gripper

The linear bushings used in vacuum were utilized as linear guides for vacuum chamber usage. All the screw holes were drilled through to avoid air pockets. A sample for test was manufactured with 200 g weight and 2 mm in diameter. The prototyping gripper was experimented to grip and hold the test sample without any failure.

CONCLUSION

Compliant mechanism could be easily applied to vacuum chamber because of the benefits with no friction and no backlash. But in most applications, the working scale only remains within nano to micrometer scale. This paper proposes a concept of flexural arms implemented three jaw parallel gripper. Three jaw grippers can center the sample automatically. Compared to conventional two jaw gripper, parallel three jaw gripper has larger operational range and more stable gripping state. The linear actuator makes the gripper capable of holding cylindrical sample sized from 1 to 3 mm in diameter. The flexural arms are the most important part that directly affect the performance of the gripper. In this paper, a kinematic model and a finite element model were built to allow different specification and requirement.

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AN APPROACH FOR DEVELOPING META MODELS OUT OF FLUID SIMULATIONS IN ENCLOSURES OF PRECISION MACHINE TOOLS

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INTRODUCTION

The sources of thermal machine tool errors are summarized by Bryan in the thermal effects diagram [1]. As shown there the temperature of machine tools electronics has an influence on the arising thermal errors and therefore on the precision of machine tools. Even though the effects are known for a long time, there is still the issue how to consider this effect in the thermal finite element (FE) simulation of machine tools.

Temperature measurements on a precision grinding machine of compact design, for low footprint, show that the temperature of the electric cabinet wall close to the machine tool bed varies between 26 °C and 40 °C during operation. Via the heat transfer mechanisms thermal conduction, radiation and convection the heat of this wall affects the temperature of the machine tool bed. Further, it affects the air between both walls, which is an enclosure, by natural convection. The temperature of the air is changing and the air starts circulating inside the enclosure. Both effects change the heat transfer coefficient (HTC) of the machine tool bed. In Figure 1, a model of the configuration is illustrated.

Considering these effects in numerical computations, computational fluid dynamics (CFD) are required. When analyzing the thermal behavior of the grinding machine using FE computations in combination with CFD the model complexity is increased, that requires high computational efforts for transient simulations. To avoid this complexity and computing time increase, only the machine tool is modeled in FE. The temperature and the HTC of the machine tool bed surface are integrated as boundary conditions. In this paper a method for deriving the correct boundary conditions at

the machine tool wall, without modeling the electric cabinet and the enclosure, is presented. A closer look on the enclosure formed between the wall of the electric control cabinet and the machine tool bed is indispensable.

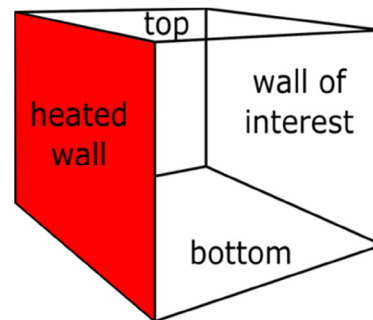


FIGURE 1. Sketch of the enclosure model. The red wall is heated to a specific temperature. The developed meta model delivers the boundary conditions at the bed wall called wall of interest.

In this study first the circulating air in the enclosure is analyzed. The enclosure is investigated by CFD and validated by measurements on an experimental setup. Afterwards a meta model describing the HTC at the wall of the machine tool bed is developed. Overall, this paper presents a general approach for developing meta models out of fluid simulations to obtain correct boundary conditions for the thermal FE simulation.

Studies on the natural convection in enclosures are done for example by Martyushev et al. [2] or Kuyper et al. [3]. Both consider a heated wall in their research. Even though this research is very helpful to get an understanding of the air convection in enclosures, it does not solve the problem of getting the HTC as a boundary condition for a FE simulation. Correct values for the HTC are of a great importance for FE simulation of machine tools as discussed by

Grossmann [4]. Neugebauer et al. [5] present a method for adapting the HTC in a time variable simulation, but mention that there is still no method available for modeling the air in enclosures. The link between the enclosure and the machine tool is missing.

FLUID SIMULATION OF ENCLOSURES

This chapter describes the simulation model of the enclosure and its parameterization. Further, model calibration via measurements is carried out. Finally the heat transfer coefficient can be evaluated.

Thermal Measurements

For validation of the fluid simulation results an experimental setup representing a cubic enclosure is used, shown in Figure 2. One vertical wall is homogeneously heated by several resistive elements arranged in a way that an almost constant temperature over the whole wall is achieved.



FIGURE 2. Experimental setup for model validation with a side length of 800 mm.

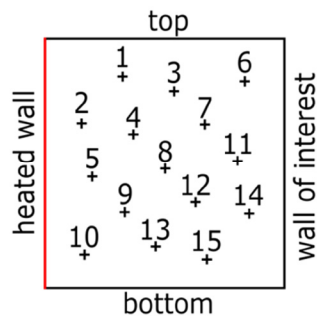


FIGURE 3. Exemplary location of temperature sensors on one side wall.

For measuring the temperature distribution 15 sensors, with a measurement uncertainty of 0.15 K, are placed on each wall. An example of the placing is shown in Figure 3. Furthermore the environmental temperature is measured with

three temperature sensors located around the experimental setup in different heights.

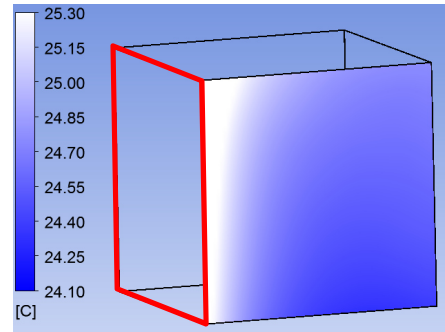


FIGURE 4. Temperature distribution on a side wall for comparison with the measurements. The red rectangle marks the connection edges to the heated wall.

The measured temperatures of all 90 sensors, 15 on every wall (including the heated wall), are used for the validation of the simulated temperatures (see Figure 4). The measurements take several hours, to make sure that the thermal steady state is reached. The comparison between the measurement and the simulations is shown exemplarily in Table 1.

Table 1. Validation of the CFD simulation with the measurement, exemplarily shown for four sensors on the side wall.

Sensor	2	5	7	15
Measurement	24.95	24.35	24.50	23.83
Simulation	24.71	24.52	24.25	24.12

Boundary Conditions

The modeled boundary conditions in CFD can be divided into two parts: the thermal conditions and the turbulence model used for the computations. The simulations are carried out as steady state simulations with different constant boundary conditions.

Thermal Conditions

The temperature of the heated wall is varied between 26 °C and 40 °C. This is equal to the temperature range of the wall of the electric cabinet measured on the machine tool. As the test stand is isolated at the bottom, the HTC of this surface is computed with the thermal conductivity λ of the isolating material and its thickness d by:

$$h = \frac{\lambda}{d} = \frac{0.028 \frac{W}{m \cdot K}}{2.5 \cdot 10^{-2} m} = 1.12 \frac{W}{m^2 \cdot K}. \quad (1)$$

The bottom temperature varies during the tests between 18 °C and 24 °C. For the outer surfaces of all other side walls and the ceiling a HTC and the environmental temperature are defined as boundary condition. The temperature of the environment varies during the test between 18 °C and 28 °C. The HTC is calculated from the environmental temperature and the temperature of the walls. Furthermore the radiation from the walls to the environment and the radiation inside the enclosure are considered in the model. The frame of the test stand where the walls are fixed is made of aluminum profiles. The measured heat conduction of the profiles is $0.3 \frac{W}{m \cdot K}$.

Turbulence Model

In the fluid model, air is considered as an ideal gas with a reference density of $1.184 \frac{kg}{m^3}$ at 20 °C. For the CFD calculation the turbulence model Speziale-Sarkar-Gatski (SSG) Reynolds stress model is used. This model is an advancement of the Reynold stress model with the approach of Speziale et al. [6]. The model solves more transport equations, while increasing the computation precision. Nevertheless, a higher calculation effort is necessary, compared to other turbulence models. The advantage of the model is that fluidic flows with a high curvature of the streamlines, which occurs in the test stand edges, can be computed with high precision and the turbulence model converges under these numerical instable conditions. The high curvature of the streamlines can be seen in Figure 5.

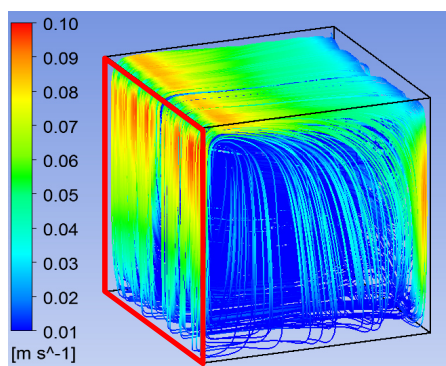


FIGURE 5. Streamlines within the enclosure when steady state is reached. The red rectangle marks the connection edges to the heated wall.

Figure 5 shows a typical air flow in the test stand of the enclosure. The air is heated at the heated

wall and climbs, due to the density reduction caused by the thermal expansion of the air. On the ceiling the air moves from the heated wall to the wall on the other side of the structure. During this movement the air is cooled. On the non-heated side walls of the structure the air is descending and further cooled. On the bottom the air movement is not directed to the heated wall. As the evaluation shows, turbulences occur and the moving of the air is no longer directed. The color map in Figure 5 shows the air velocity. It can be seen that the highest velocities occur on the heated wall and in the corners of the opposite wall. A cut through the enclosure, shown in Figure 6, illustrates that high velocities just occur near the walls. In the center of the enclosure there is a huge area where almost no air movement occurs.

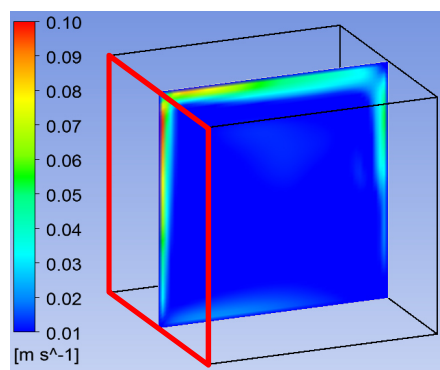


FIGURE 6. Cut through the enclosure. The red rectangle marks the connection edges to the heated wall.

All of the rising air flows along the ceiling, distributed to the surfaces of the side walls. The cooled air partially flows from the side walls back to the heated wall, whereby the mass flow at the bottom is lower than at the ceiling. These results underline the necessity of having a close look on the circulating air in the enclosure.

Evaluation of the Heat Transfer Coefficient

The simulation results deliver the wall HTC of the inner surface of the enclosure. An averaged HTC is taken as the output of the simulation for the wall of interest. The HTC changes, depending on the three temperatures of the:

- heated wall: T_h ,
- environmental: T_e and
- bottom: T_b .

Figure 7 shows the HTC distribution at the inner surfaces of the model. The red rectangle marks the connection edges of the heated wall for a better orientation. It can be seen, that the HTC is the highest above the heated wall. This is due to the fact that in this area the inner air temperature is the highest; consequently also the temperature difference between the inner air temperature and the environmental temperature is the highest. Additionally in this region the air has the highest velocity.

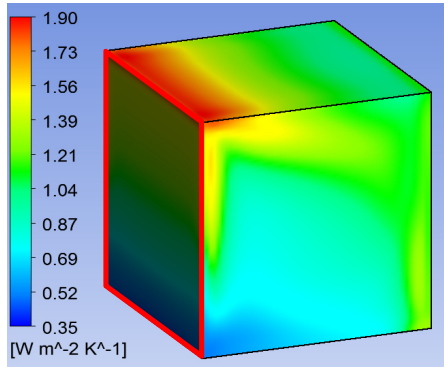


FIGURE 7. Wall heat transfer coefficient in steady state at inner model surfaces. The red rectangle marks the connection edges to the heated wall. Results are shown for $T_h = 38^\circ\text{C}$, $T_e = 24^\circ\text{C}$ and $T_b = 22^\circ\text{C}$.

Figure 8 shows the HTC on the wall of interest.

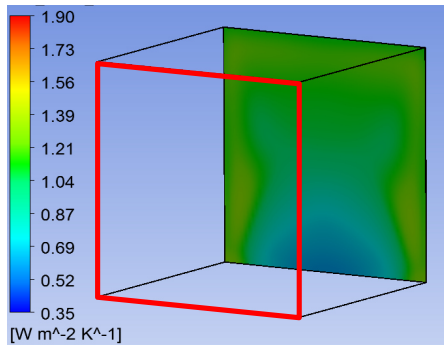


FIGURE 8. Wall heat transfer coefficient in the steady state. The red rectangle marks the connection edges to the heated wall. Results are shown for $T_h = 38^\circ\text{C}$, $T_e = 24^\circ\text{C}$ and $T_b = 22^\circ\text{C}$.

It can be seen, that on this wall the range of the HTC is not as large as on the top wall. An averaged HTC for the derived meta model is thus expected to give a good estimation of the HTC. In this case the averaged HTC for the wall is $1.06 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$. This value is then taken for

developing the meta model as it is explained in the following section.

For all other walls the explained method works as well. Exceptionally for the top wall it has to be considered that the uncertainty is larger as the HTC varies more there and the model just gives a single value for the whole ceiling. If the ceiling surface is connected to the machine tool structure, it is advisable to discretize the ceiling surface into several regions having a lower variation of the HTC.

META MODEL

This section first describes the development of the meta model for enclosures. Afterwards an interpretation with two examples and the validation of the meta model is discussed.

Development of the Meta Model

To get a broad data basis for deriving the meta model the temperatures T_h , T_e and T_b of the CFD simulation model are varied 40 times. The results of these 40 models are the basis for the meta model. For describing the HTC h a numeric value equation with linear and quadratic terms can be derived:

$$h = b_1 + b_2 \cdot T_h + b_4 \cdot T_e + b_6 T_b + b_3 \cdot T_h^2 + b_5 \cdot T_e^2 + b_7 \cdot T_b^2 + b_8 \cdot T_h \cdot T_e + b_9 \cdot T_h \cdot T_b + b_{10} \cdot T_e \cdot T_b \quad (2)$$

In (2) b_1 is a zero balancing factor. The influence of the three temperatures T_h , T_e and T_b are considered with linear terms and with polynomials of second order in accordance to the theory of convective heat transfer. Furthermore a combination of two of them is considered with respect to the radiation. The model terms are weighted with constant parameters, computed using least squares. The values for the constant parameters b_1 to b_{10} are listed in Table 2.

Table 2. Coefficients for computing the HTC with the meta model at the wall of interest.

b_1	b_2	b_3	b_4	b_5
16.7	0.83	$-2.93 \cdot 10^{-3}$	-1.0	$-2.02 \cdot 10^{-3}$
b_6	b_7	b_8	b_9	b_{10}
0.04	$-2.76 \cdot 10^{-4}$	$4.91 \cdot 10^{-3}$	$-1.55 \cdot 10^{-3}$	$2.11 \cdot 10^{-3}$

Interpretation of the Meta Model

In Figure 9 the HTC for the wall of interest is shown as a function of the environmental temperature and the temperature of the heated wall. The temperature of the bottom is kept constant to 22 °C.

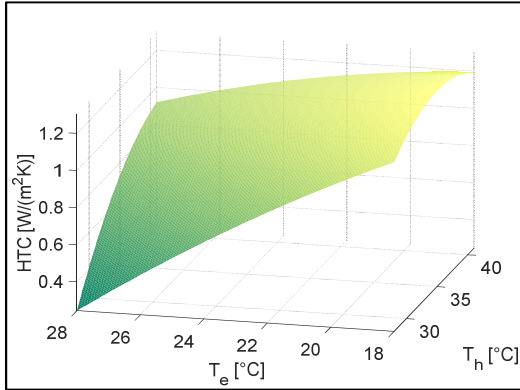


FIGURE 9. Wall heat transfer coefficient HTC on the wall of interest calculated with the meta model; bottom temperature 22°C.

The results in Figure 9 illustrate on the correlation between T_h and T_e that low temperature differences slightly influence the HTC what allows to use one single value for the whole wall of interest. It can be seen that the correlation between the HTC and the environmental temperature is almost linear. For the case of the correlation between the heated wall and the HTC this is not so obvious. A simplification to a linear meta model would thus have a larger error in calculating the HTC.

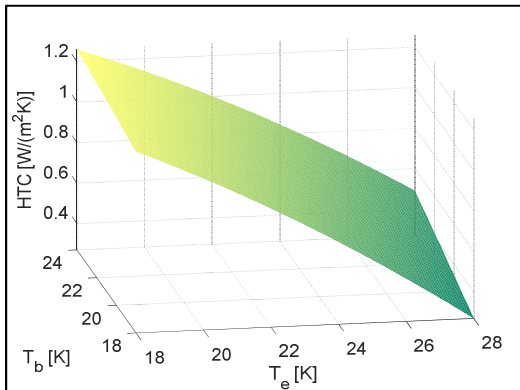


FIGURE 10. Wall heat transfer coefficient on the wall of interest calculated with the meta model; heated wall temperature 30 °C.

In Figure 10 the temperature of the heated wall is kept constant at 30 °C. The HTC is shown as

a function of the environmental temperature and the bottom temperature. In both cases the linear correlation is obvious. It can be seen that the temperature of the bottom has a lower influence on the HTC, compared to the environmental temperature, which is caused by the arising flow field illustrated in Figure 5.

Validation of the Meta Model

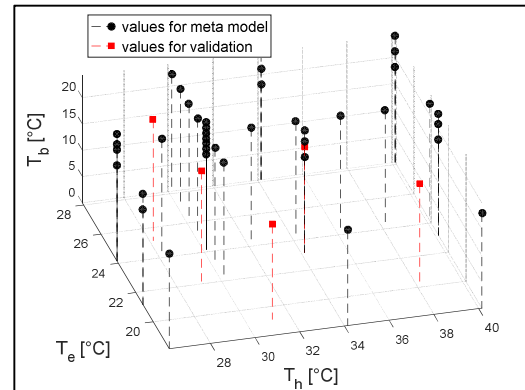


FIGURE 11. Validation of the meta model.

To validate the model, five additional CFD simulations are performed. They are shown by the red squares in Figure 11. The black dots show the temperatures for the 40 fluid simulations used for deriving the meta model. These five additional models are used to test the validity of the meta model.

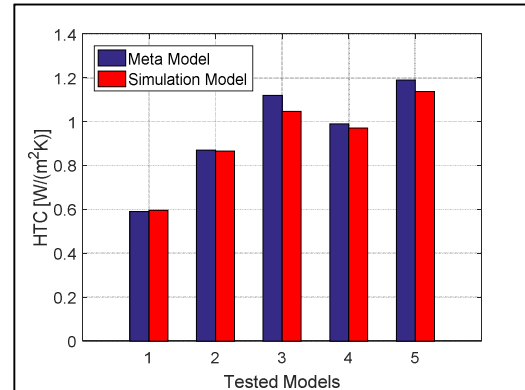


FIGURE 12. Comparison between the meta model output and simulation results for the five additional models.

The HTC given from the five CFD simulations are compared with the HTC obtained from the meta model for those five parameter sets. The results can be seen in Figure 12. The maximum difference is $h = 0.07 \frac{W}{m^2K}$. This shows, that the

meta model fits very well within the defined parameter range.

CONCLUSIONS

In this paper an investigation of the HTC on enclosures of machine tools is presented. The approach of investigation is further used to derive a meta model for the description of the HTC acting on the machine tool surface. Due to the simple structure of the developed meta model, the HTC valid for the bed surface of the machine tool the model can be implemented in the FEM computations for investigating the thermal behavior of the machine. As the model considers temperature boundary conditions (T_h , T_e and T_b), which can be easily measured, it makes it possible to find the correct HTC very efficiently. This becomes very important when a high precision machine tool is simulated and the computations are compared to measurements for evaluation, as the change of boundary conditions can be considered without increasing the computational cost. The paper presents a general approach how meta models for HTC can be developed. This makes the approach usable for application on other precise machines, for example for metrological devices such as coordinate measuring machines.

FUTURE WORK

The presented method for developing meta models for enclosures is of general use. Therefore it will be used to develop meta models for further shapes of enclosures. One of the most important fields of application is enclosures in machine tools, because under every machine housing an enclosure between the housing and the structure of the machine tool can be found. As Mayr et al. [7] discuss the main error sources in production are caused by changing ambient temperatures. Meta models with respect to these influences will be developed with the approach. This leads to a large number of different sizes of enclosures. The future challenge is how meta models can be parametrized for different sizes of machine tool enclosures. Here an analysis of non-dimensional numbers will be important.

In Table 2 it can be seen, that besides the constant factor b_1 the weighting factors of the linear terms have the highest influence on the HTC. This leads to the question whether linear models would deliver the requested accuracy. Moreover the coefficients of the meta model show that the influence of the bottom

temperature is more than one order of magnitude lower as the one of the temperature of the heated wall and the environment. Investigations on simplifications of the model by neglecting quadratic terms and the bottom temperature changes will be carried out, with respect to the changes in computation of the thermal machine tool errors evaluated at the TCP.

ACKNOWLEDGEMENT

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Machine tool oriented optimization of 3-RPS parallel manipulator for rotation and uniform deflecting abilities maximization

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INSTRUCTIONS

Parallel mechanisms have been successfully used as high-precision motion stages in machine tools. Generally, these parallel machine tools have the conveniences of high rigidity, low inertia, and high dynamics, whereas they are also suffering limited Cartesian workspace and incompetent orientation workspace [1]. In order to overcome the defect of small Cartesian workspace, usually a hybrid kinematic structure is applied. Another challenging task is to enlarge the orientation workspace or the tilting angle by optimizing the structural parameters. Moreover, the uniform deflecting abilities of a parallel machine tool, which act as important criteria to realize the NC machining, should be involved in. In the practical design, discrete distribution of the structural parameters is also an inevitable factor influences the optimization results.

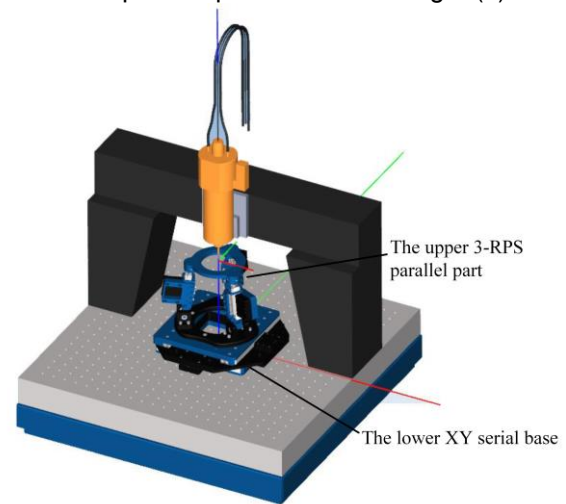
Amounts of researches have been implemented to optimize the structural parameters of a parallel machine tool. Among them, the objective function based optimization, is an efficiently and widely used method [2, 3]. Unfortunately, most researches only concern about the tilting angle and use approximately matched parameters in optimization, which may deteriorate the machining ability of the machine tool.

This article proposes an orientation workspace optimization algorithm that can automatically search for a set of optimal structural parameters for designing a parallel mechanism. A general type of 3-RPS spatial parallel mechanism [4] is considered as the study object, which is adopted as a parallel module in a micro machine tool.

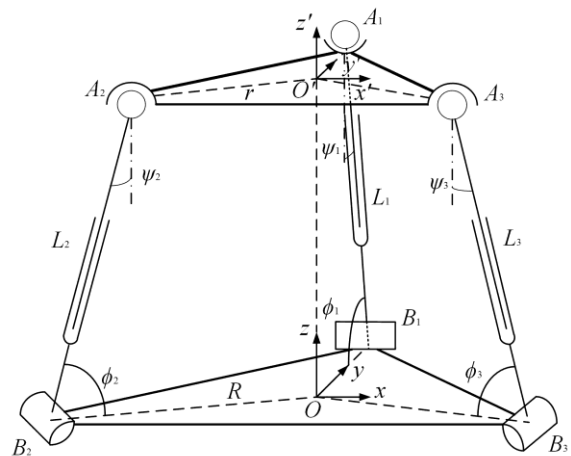
Firstly, with a proposed XY-RPS (3T2R) five-axis machine tool, a new definition of the orientation workspace is presented. Secondly, a uniform deflection factor is introduced for justifying the consistence of the evaluated deflecting abilities. Then, the effects of the design parameters on the rotational capacity are also investigated. Finally, a modified NSGA-II algorithm is employed for the optimization, whose results are conducted to prototype design.

FIVE-AXIS MICRO MACHINE TOOL

A RPS-XY TRIPOD motion stage is suggested to use in a five-axis micro machine tool, which is illustrated in Fig. 1(a). Targeting for high-precision micro-machining, the stage is actuated by linear ultrasonic motors and constructed by high-precision components. The schematic of the 3-RPS parallel part is shown in Fig. 1(b).



(a) The proposed five-axis micro machine tool



(b) The schematic of 3-RPS parallel mechanism

FIGURE 1. Schematics of the micro machine tool and 3-RPS parallel mechanism

Lee and Shah [5] studied the inverse kinematics of 3-RPS, in which the link lengths can be

obtained by z-y-z Euler angles (α , β , γ) with analytical solutions.

ORIENTATION WORKSPACE

With $(x_c, y_c, z_c)^T$ denoting the origin of the moving platform, the workspace is defined by mapping from the independent variables (α , β , z_c) to cylindrical coordinates (ρ , ϕ , z) in Fig. 2.

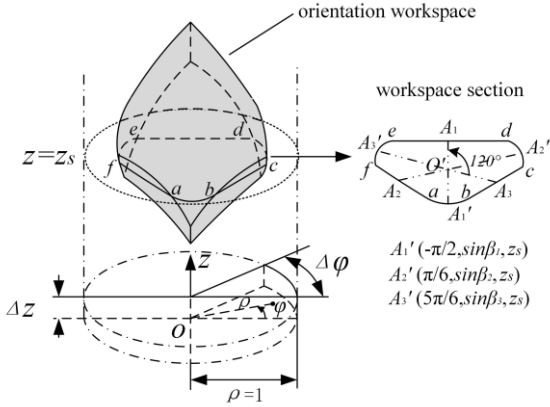
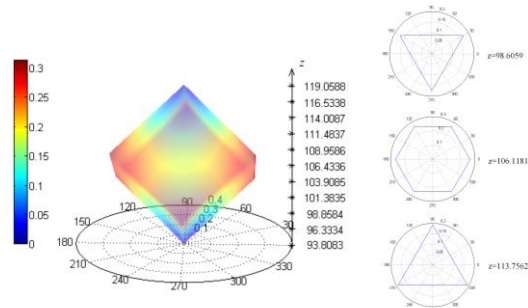


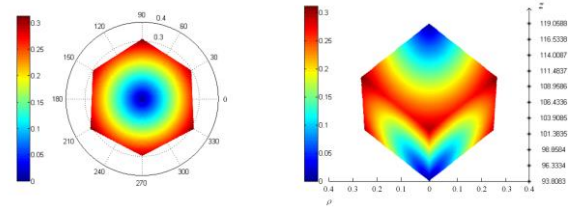
FIGURE 2. Orientation workspace definition and workspace section analysis

Here the orientation workspace is calculated based on a combination of numerical and analytical methods. A series of sub-workspaces parallel to Opφ plane with a set step size Δz are combined to form the whole workspace. By using inverse kinematics, a dichotomy-based searching method is adopted to determine the boundary of each sub-workspace.

Fig. 3 demonstrates the calculated workspace (3(a)), corresponding projections onto the Opφ plane (3(b)) and the Opz plane (3(c)) for an instanced 3-RPS stage.



(a) Orientation workspace and workspace sections



(b) Projection onto Opφ (c) Projection onto Opz

FIGURE 3. Orientation workspace specification

UNIFORM DEFLECTION FACTOR

In order to realize five-axis NC machining, while the workpiece reaches a point P in the Cartesian workspace, its tilt angle can randomly change in the conic face of β , but not interfere. Actually, the consistency of the all-direction deflecting abilities ensures the continuity in the change of the workpiece's tilt angle with respect to direction angles. And a high consistency can enlarge the conic face of β , which facilitates complex surface machining. The schematic can be illustrated in Fig. 4.

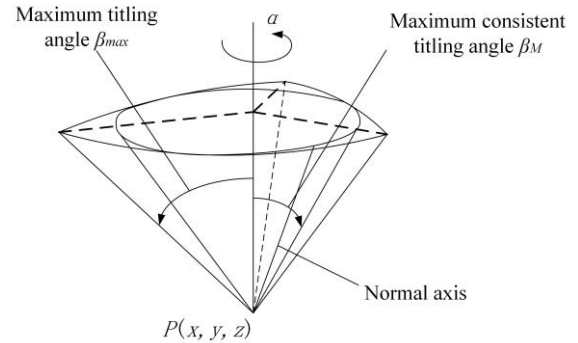


FIGURE 4 Orientation of the normal axis of workpiece

The uniform deflection factor δ is defined as the ratio of the volume of the uniform deflection workspace to that of the whole workspace. Accordingly, the maximum tilting angle and the uniform deflection factor are determined as the two main indexes to indicate the machining capability of a parallel mechanism.

CONTINUOUS AND DISCRETE DESIGN PARAMETERS

The engineering assembly model of 3-RPS parallel mechanism illustrated in Fig. 5. And the relevant specifications and parameters are given in Fig. 6 and Fig. 7.

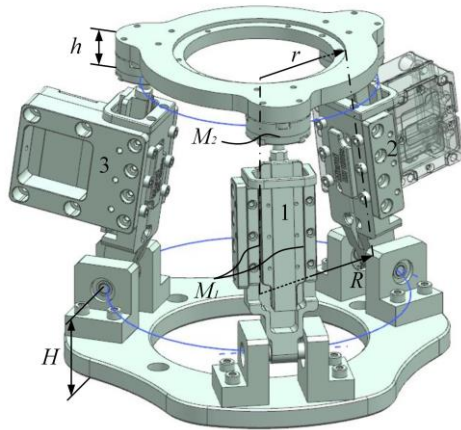


FIGURE 5 Engineering assembly model of 3-RPS parallel mechanism

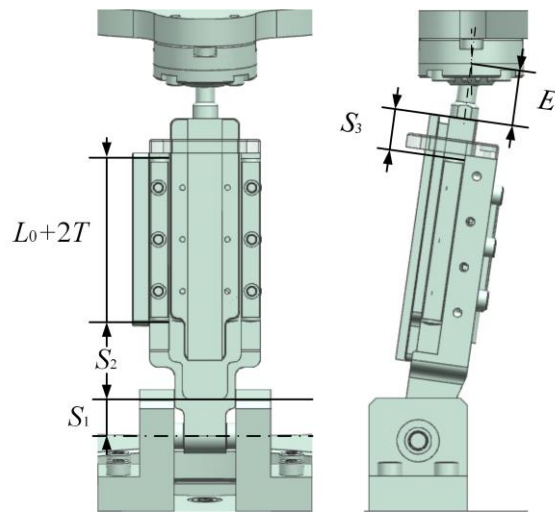
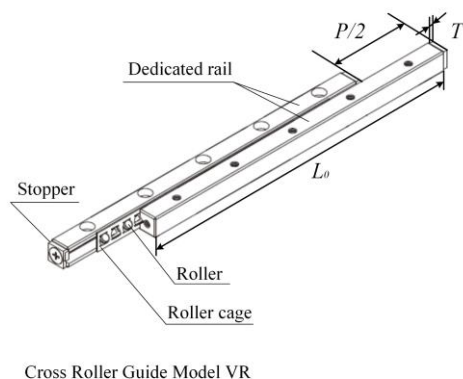
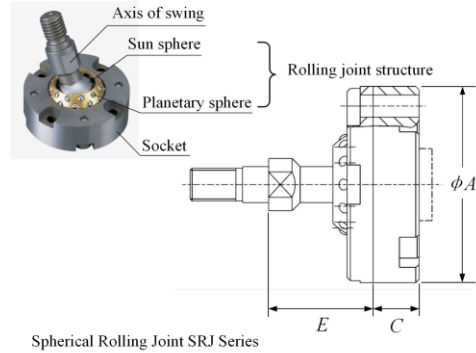


FIGURE 6 Configuration of a link



(a) Linear guide specification



(b) Spherical joint specification

FIGURE 7 Commercial products specification

The parameters can be sorted into two categories, the continuous ones of customizable design parameters and the discrete ones of un-customizable design parameters.

The continuous design parameters include the radius of the moving platform r , the radius of the base platform R , the height of the moving platform h , the height of the base platform H , the mounting angle of the spherical joint ζ , and the customizable lengths of the link S_1 , S_2 , S_3 .

The discrete design parameters are given by un-designable standard components, i.e., the linear guide and the spherical joint.

EFFECTS ON THE ROTATIONAL CAPACITY

(1) Ratio of the moving platform radius to the base platform radius λ

The maximum tilting angle has a general trend of decreasing with λ .

(2) Heights of the moving and base platforms

Both of them have no effect on the rotational capacity of the stage.

(3) Mounting angle of the spherical joint

A variation in the mounting angle may result in a change of rotational capacity.

(4) Product models of the linear guide and the spherical joint

Obviously, the product models M_1 and M_2 decide the link length and the limit range of angle of swing, which affect the rotational capacity.

OPTIMIZATION

To obtain the non-dominated solutions of the case, the NSGA-II algorithm proposed by Deb et al. [6], which is the most preferred algorithm to solve the multi-objective problems based on the principles of GA, is adopted. However, for the case in this study, the standard algorithm is appropriately modified.

The five-dimension vector of decision variables is given as $(\lambda, S, \zeta, M_1, M_2)^T$, in which S denotes the addition of S_1, S_2 and S_3 .

The GA parameters and the corresponding modifications are

- Chromosome representation: The gray representation;
- Selection function: The tournament selection process with crowded-comparison-operator;
- Genetic operators: Fixed-probability uniform crossover and simple mutation with the crossover rate 0.9 and the mutation rate 0.1;
- Population size: 20;
- Evaluation function: Each evaluation function consists of two parts, an objective function modeled by a multi-level fuzzy evaluation method with considering other criteria, and an adaptive penalty function balancing the objective function and the constraint violations

An analytic hierarchy process (AHP) with 9-unit scales [7] is adopted to produce the objective functions, which considers the radius ratio λ and the motion range of the passive revolute joint $\Delta\Phi$.

Each adaptive penalty function is constructed referring to Liu's work [8].

The optimization is formulated to find the solutions of large titling angel and uniform deflecting factor, and to ensure that the moving platform cannot be too small and $\Delta\Phi$ be small.

With the number of iterations set to 100 the Pareto front is shown in Fig. 8. Note that f_{p1} and f_{p2} denote the two evaluation functions. Surely there is no single solution for which all criteria are optimal simultaneously, and each solution is important with respect to some trade-off relationship among the criteria.

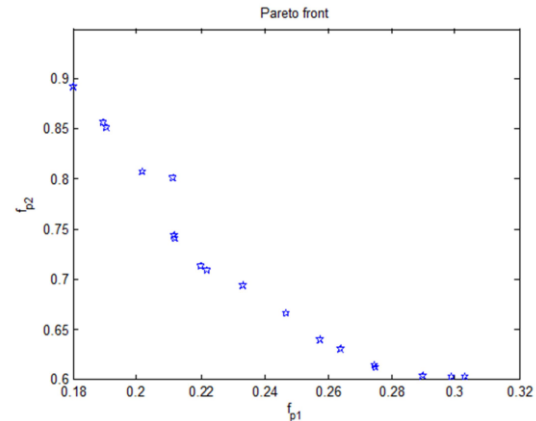
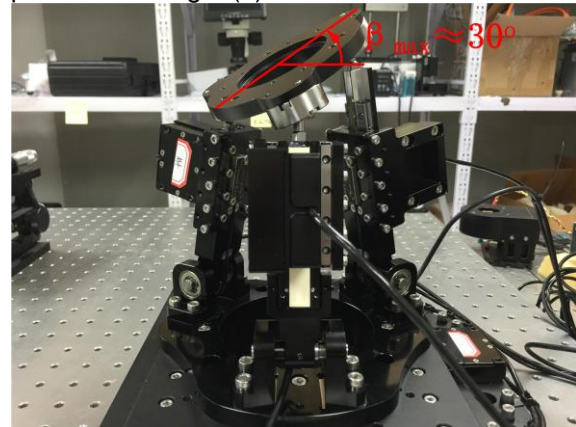


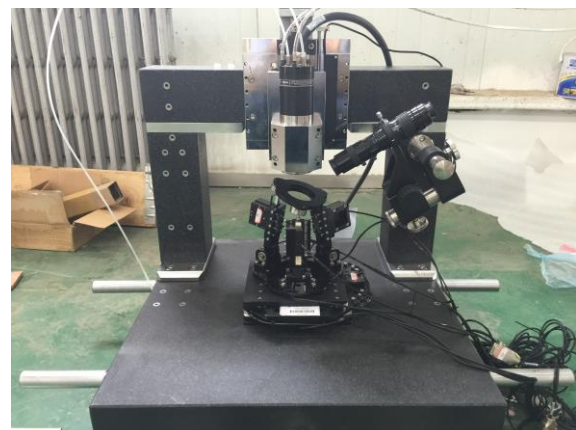
FIGURE 8 Pareto optimal front

PROTOTYPE OF TRIPOD AND MICRO MACHINE TOOL

The designed stage, which has good rotational capacity ($\beta_{\max} \approx 30^\circ$ and $\delta \approx 0.80$), is shown at its maximum titling pose in Fig. 9(a). And prototype of micro machine tool is developed and presented in Fig. 9(b).



(a) Prototype of TRIPOD



(b) Prototype of micro machine tool

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Approximate Formulas for the Motion Errors of Hydrostatic Guideways under Quasi-static Condition

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INTRODUCTION

Fluid hydrostatic guideways are widely used in precision, ultra-precision processing technology for their inherent high rigidity, high precision, and low friction. Their high precision comes from the error averaging effect of the high pressure oil film [1]. Therefore, many experts and scholars have attempted to explain this phenomenon.

Shamoto et al. got the transfer function between the geometric errors of the guide rails and motion errors of the table using the finite element method (FEM) [2]. Wang et al. predicted the influence of speed on motion errors of hydrostatic guideways by solving the Reynolds equation, flow continuous equation and motion equation simultaneously [3]. Qi et al. analyzed the influence of the longitudinal harmonic errors of the rails on motion errors of hydrostatic guideways using the equivalent film thickness [4]. Kashchenevsky et al. got the approximate formulas for motion errors of thrust bearings when the disk and bush have geometric errors using the equivalent third power of film thickness [5].

In order to reduce the difficulty of calculation, we derive the new approximate formulas for the motion errors of hydrostatic guideways using the average third power of the film thickness based on the principle that the fluid resistance of the land is equal under quasi-static condition in this article. Furthermore, we get the approximate formulas for averaging effect coefficients of vertical displacement and yaw angle. Finally, we make a discussion about the influence of the land width and exhausted groove width on averaging effect based on the formulas.

THEORY AND CALCULATION

Structure of the hydrostatic guideways

As shown in Figure 1 and Figure 2, the table has four pads, of which two are on the upside and two are on the downside. The restrictors used

for all pads are fixed resistance restrictors, and the geometric errors of the guide rails can be decomposed into harmonic waves of different frequencies. While symbols B , L and h_0 are the width, height and initial film thickness of the pads, respectively, symbols a , b and c are the lateral land width, longitudinal land width and exhausted groove width of the pads, respectively. Additionally, the coordinate systems are illustrated in Figure 1 and Figure 2, while the positive direction of yaw angle β is around y -axis.

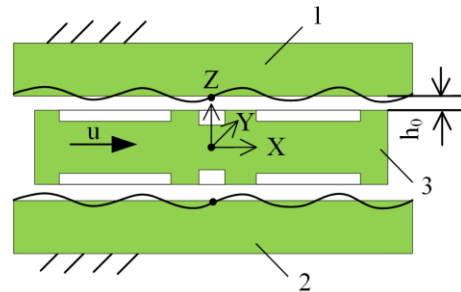


FIGURE 1. The structure diagram of hydrostatic guideways. 1-upside guide rail, 2-downside guide rail, 3-table.

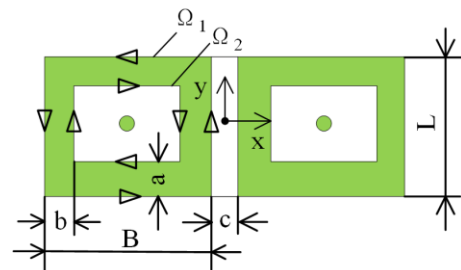


FIGURE 2. The structure diagram of the pads.

Flow continuous equation

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Initially, when the table is at the equilibrium position, the flow continuous equation is as follows:

$$\bar{B} \frac{h_0^3}{\eta} p_{r0} = \frac{1}{R_c} (p_s - p_{r0}) \quad (1)$$

Where the structural parameter $\bar{B}$ equals to $[(B-b)b + a(L-a)]/6ab$. Additionally, while p_{r0} is the initial pressure of the recess and p_s is the supply pressure of the inlet, η and R_c are the dynamic viscosity and fixed resistance of the restrictor, respectively.

After the table travels to a new position and is rebalanced, we can obtain:

$$\bar{B} \frac{h_e^3}{\eta} p_r = \frac{1}{R_c} (p_s - p_r) \quad (2)$$

Where p_r is the rebalanced pressure of the recess, and h_e^3 is the equivalent third power of film thickness based on the principle that the fluid resistance of the land is equal.

Combining Equation (1) with Equation (2), we can get:

$$p_r = \frac{zh_0^3}{zh_0^3 + h_e^3} p_s \quad (3)$$

Where β equals to p_{r0}/p_s and z equals to $\beta/(1-\beta)$.

Quasi-static equilibrium equation

When the moving velocity is very slow which means the table is under quasi-static condition, we obtain:

$$\begin{cases} (p_{r1L} + p_{r1R}) A_e = (p_{r2L} + p_{r2R}) A_e \\ (p_{r2L} - p_{r1L}) A_e \frac{B+c}{2} = (p_{r2R} - p_{r1R}) A_e \frac{B+c}{2} \end{cases} \quad (4)$$

Where A_e is the effective loading area of a single pad, and p_{r1L} , p_{r2L} , p_{r1R} , p_{r2R} are the pressures of the left upside pad, left downside pad, right upside pad, right downside pad, respectively.

$$\begin{cases} h_{1eL}^3 = h_{2eL}^3 \\ h_{1eR}^3 = h_{2eR}^3 \end{cases} \quad (5)$$

The Equation (5) can be derived from Equation (4), where h_{1eL}^3 , h_{2eL}^3 , h_{1eR}^3 and h_{2eR}^3 are the equivalent third power of film thickness of the left upside pad, left downside pad, right upside pad and right downside pad, respectively.

Equivalent third power of the film thickness

The film thickness expressions of the upside guide rail h_1 and downside rail h_2 are:

$$\begin{cases} h_1 = h_0 - Z_0 + E_1 \sin\left(\frac{2\pi n_1}{B+c} X + \varphi_1\right) + \beta_0 X \\ h_2 = h_0 + Z_0 - E_2 \sin\left(\frac{2\pi n_2}{B+c} X + \varphi_2\right) - \beta_0 X \end{cases} \quad (6)$$

Where Z_0 is the vertical displacement and β_0 is the yaw angle. While n_1 and n_2 are the numbers of the waves on the upside and downside guide rails in the space of two adjacent pads, respectively, E_1 , φ_1 and E_2 , φ_2 are the amplitudes and phase angles of the harmonic waves on the upside and downside rails, respectively.

Based on the Taylor formula, the third power of change ratio of the film thickness can be rewritten into an approximate equation:

$$\left(1 + \frac{\Delta h}{h_0}\right)^3 = 1 + 3\frac{\Delta h}{h_0} + 3(1+\theta)\left(\frac{\Delta h}{h_0}\right)^2 \approx 1 + 3\frac{\Delta h}{h_0} \quad (0 < \theta < 1) \quad (7)$$

Where symbol $\Delta h/h_0$ is the change ratio of film thickness. When $\Delta h/h_0$ is less than 10%, the calculation error of $(1 + \Delta h/h_0)^3$ is less than 8.3%.

So we can get:

$$\begin{cases} h_1^3 \approx h_{1-3} = h_0^3 \left[h_0 - 3Z_0 + 3E_1 \sin\left(\frac{2\pi n_1}{B+c} X + \varphi_1\right) + 3\beta_0 X \right] \\ h_2^3 \approx h_{2-3} = h_0^3 \left[h_0 + 3Z_0 - 3E_2 \sin\left(\frac{2\pi n_2}{B+c} X + \varphi_2\right) - 3\beta_0 X \right] \end{cases} \quad (8)$$

The equivalent third power of the film thickness is shown in the Equation (9) based on the principle that the fluid resistance of the land is equal.

$$h_e^3 = \frac{1}{A} \iint_{Land} h^3 dx dy \quad (9)$$

Where A is the area of land of single pad which equals to $2[a(B-b) + b(L-a)]$.

Based on the mathematical Green formula, we can change the surface integration into the boundary integration for each pad, which greatly simplifies the calculation.

$$h_e^3 = \frac{1}{A} \iint_{Land} h^3 dx dy = -\frac{1}{A} \oint_{Edges} h^3 y dx \quad (10)$$

The directions of the edges Ω_1 and Ω_2 are shown in Figure 2 for one pad. After the integration for each pad, the equivalent third power of the film thickness can be expressed as:

$$\begin{cases} h_{0L}^3 = \frac{h_0^2}{A} \left[(h_0 - 3Z_0)K_3 + 3E_1 \sin(\varphi_1 - \pi n_1)K_1 - \frac{3}{2}\beta_0(B+c)K_3 \right] \\ h_{20L}^3 = \frac{h_0^2}{A} \left[(h_0 + 3Z_0)K_3 - 3E_2 \sin(\varphi_2 - \pi n_2)K_2 + \frac{3}{2}\beta_0(B+c)K_3 \right] \\ h_{0R}^3 = \frac{h_0^2}{A} \left[(h_0 - 3Z_0)K_3 + 3E_1 \sin(\varphi_1 + \pi n_1)K_1 + \frac{3}{2}\beta_0(B+c)K_3 \right] \\ h_{20R}^3 = \frac{h_0^2}{A} \left[(h_0 + 3Z_0)K_3 - 3E_2 \sin(\varphi_2 + \pi n_2)K_2 - \frac{3}{2}\beta_0(B+c)K_3 \right] \end{cases} \quad (11)$$

Where

$$\begin{cases} K_1 = \frac{(B+c)}{\pi n_1} \left[L \sin\left(\pi n_1 \frac{B}{B+c}\right) - (L-2a) \sin\left(\pi n_1 \frac{B-2b}{B+c}\right) \right] \\ K_2 = \frac{(B+c)}{\pi n_2} \left[L \sin\left(\pi n_2 \frac{B}{B+c}\right) - (L-2a) \sin\left(\pi n_2 \frac{B-2b}{B+c}\right) \right] \\ K_3 = 2[a(B-b) + b(L-a)] \end{cases} \quad (12)$$

Approximate formulas for the motion errors

Combine the Equation (11) with Equation (5) and we can get the vertical displacement Z_0 and yaw angle β_0 :

$$\begin{cases} Z_0 = \frac{1}{2K_3} [K_1 E_1 \sin(\varphi_1) \cos(\pi n_1) + K_2 E_2 \sin(\varphi_2) \cos(\pi n_2)] \\ \beta_0 = -\frac{1}{(B+c)K_3} [K_1 E_1 \cos(\varphi_1) \sin(\pi n_1) + K_2 E_2 \cos(\varphi_2) \sin(\pi n_2)] \end{cases} \quad (13)$$

Where:

$$\begin{cases} \varphi_1 = ut \frac{2\pi n_1}{(B+c)} + \varphi_{10} \\ \varphi_2 = ut \frac{2\pi n_2}{(B+c)} + \varphi_{20} \end{cases} \quad (14)$$

Where u is the velocity and t is the time. The symbols φ_{10} and φ_{20} are the initial phase angles of the upside and downside harmonic waves, respectively.

The equation (13) has two parts. The left part containing n_1 and the right part containing n_2 are the motion errors caused by the harmonic errors of the upside and downside guide rails, respectively. Additionally, the items $\sin(\varphi_1)$ and $\sin(\varphi_2)$ are the coefficients of motion errors caused by the upside and downside guide rails, respectively. When the moving distance of the table is far enough, the absolute values of $\sin(\varphi_1)$ and $\sin(\varphi_2)$ can both reach maximums. Form Equation (14), the spatial periods of coefficients $\sin(\varphi_1)$ and $\sin(\varphi_2)$ can be obtained, of which the values are $(B+c)/n_1$ and $(B+c)/n_2$, respectively.

When only the upside guide rail has geometric errors, we define the averaging effect coefficients as follows:

$$\begin{cases} \delta = \frac{Z_0}{E_1} = \left| \frac{K_1 \cos(\pi n_1)}{2K_3} \right| \\ \gamma = \frac{\beta_0(B+c)}{E_1} = \left| \frac{K_1 \sin(\pi n_1)}{K_3} \right| \end{cases} \quad (15)$$

The symbol δ is the averaging coefficient of straightness errors, which represents the ratio between amplitude of the vertical displacement and amplitude of the harmonic wave on the rail. Additionally, the symbol γ is the averaging effect coefficient of yaw errors, which represents the ratio between amplitude of the yaw angle and the value $E_1/(B+c)$.

RESULTS AND DISCUSSIONS

Discussion of the approximate formulas

Some conclusions can be easily obtained from Equation (13):

(1) The overall motion errors can be obtained from the linear superposition of the part caused by the harmonic wave errors of the upside guide rail and the part caused by the harmonic wave errors of the downside guide rail. Additionally, the periods during which the vertical displacement and yaw angle change with the moving distance equal to the wavelength of the harmonic waves.

(2) On the condition that the amplitudes of the harmonic waves on the upside and downside guide rails are equal, when the two phase angles of them are reversed the vertical displacement and yaw angle are zero, and when the two phase angles of them are equal the vertical displacement and yaw angle double compared to the situation that just one side of the guide rails has geometric errors.

Furthermore, some conclusions can be easily obtained from Equation (15):

(1) When there are $n_1=0.5, 1.5, 2.5, 3.5\ldots$ waves on the guide rails in the space of two adjacent pads, the straightness errors are zero and when there are $n_1=1, 2, 3, 4\ldots$ waves on the guide rails in the space of two adjacent pads, the yaw errors are zero.

(2) When the wave number n_1 is very small which means the wavelength is infinite, the averaging coefficient δ of straightness errors is 0.5, which can be obtained from the calculation of the limit of the Equation (15). However, due to the fact that the moving distance is limited which means the item $\sin(\varphi)$ is zero, the actual vertical displacement Z_0 is zero according to the Equation (13).

Vertical displacement and yaw angle change with the moving distance

In the following calculation, the default parameters are used: $B=60.5\text{mm}$, $L=50\text{mm}$, $h_0=20\mu\text{m}$. Figure 3 and Figure 4 are calculated based on the condition that the amplitudes and phase angles of the harmonic waves on the upside and downside are equal. The amplitudes of harmonic waves used here are $2.5\mu\text{m}$ and the moving distance of the table is 258mm (four times the space of adjacent pads $4(B+c)$).

From Figure 3, we can know, when there exists 0.5 harmonic wave on the guide rail in the space of two adjacent pads the vertical displacement is zero, and when there exists 0.75 harmonic wave the vertical displacement can reach the maximum $0.4\mu\text{m}$. From Figure 4, we can know, when there exists 0.5 harmonic wave on the guide rail in the space of two adjacent pads, the yaw angle can reach the maximum $10''$, and when there exists 1 harmonic wave the yaw angle is zero. Furthermore, the number of fluctuations of vertical displacement and yaw angle is equal to the number of harmonic waves in the space of two adjacent pads.

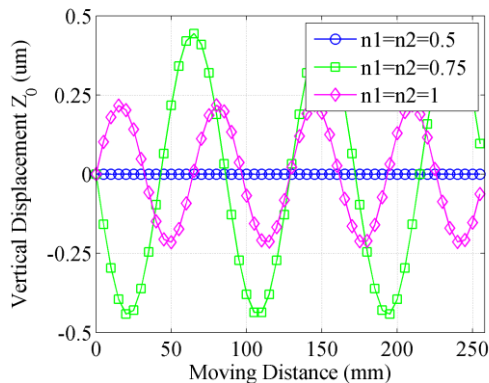


FIGURE 3. The vertical displacement changes with the moving distance, when $a=b=15\text{mm}$, $c=4\text{mm}$, $E_1=E_2=2.5\mu\text{m}$.

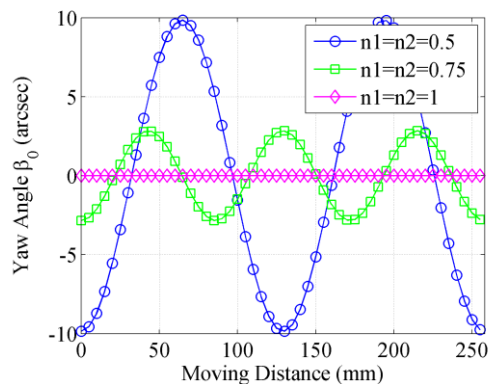


FIGURE 4. The yaw angle changes with the moving distance, when $a=b=15\text{mm}$, $c=4\text{mm}$, $E_1=E_2=2.5\mu\text{m}$.

Influence of land width on the straightness errors and yaw errors

As shown in Figure 5, when the number of harmonic waves on the guide rail in the space of two adjacent pads is smaller than 1, the straightness averaging coefficient increases with the growth of land width, meaning the reduction of averaging effect. Moreover, when the number of harmonic waves is greater than 1, the straightness averaging coefficient decreases with the growth of land width, meaning the enhancement of averaging effect.

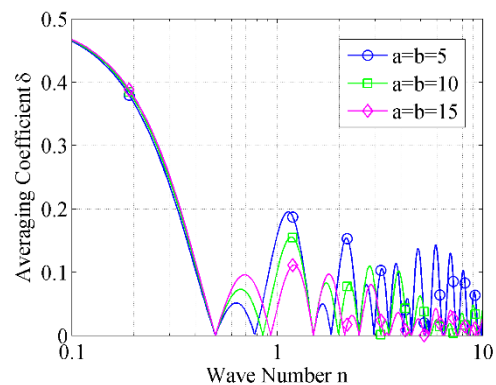


FIGURE 5. The averaging coefficient of straightness errors changes with the land width, when $c=4\text{mm}$.

As shown in Figure 6, when the number of harmonic waves on the guide rail in the space of two adjacent pads is smaller than 1, the yaw averaging coefficient increases with the growth of land width, meaning the reduction of averaging effect. Moreover, when the number of harmonic waves is greater than 1, the yaw averaging coefficient decreases with the growth of land width, meaning the enhancement of averaging effect.

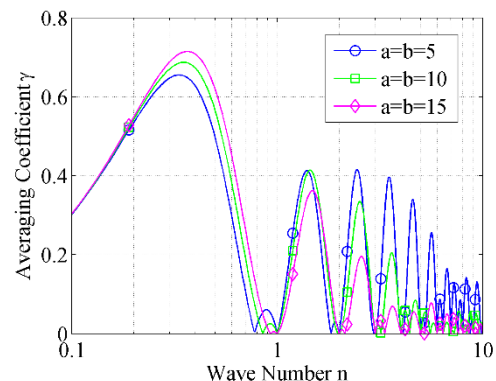


FIGURE 6. The averaging coefficient of yaw errors changes with the land width, when $c=4\text{mm}$.

Influence of exhausted groove width on the straightness errors and yaw errors

Figure 7 shows the straightness averaging coefficient increases with the growth of exhausted groove width which means the reduction of averaging effect.

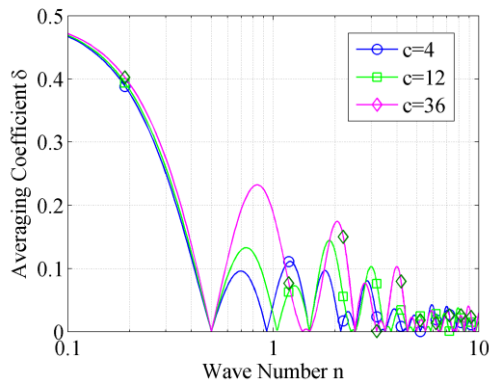


FIGURE 7. The averaging coefficient of straightness errors changes with the exhausted groove width, when $a=b=15\text{mm}$.

Figure 8 shows the yaw averaging coefficient increases with the growth of exhausted groove width which means the reduction of averaging effect.

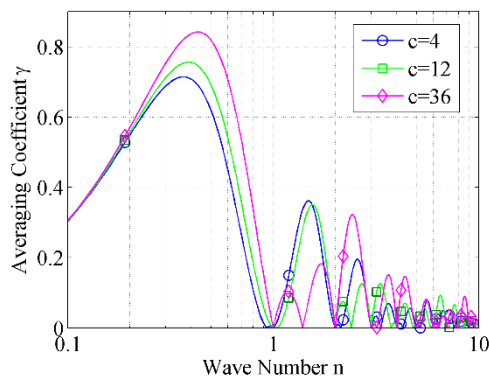


FIGURE 8. The averaging coefficient of yaw errors changes with the exhausted groove width, when $a=b=15\text{mm}$.

It is worth noting that the growth of exhausted groove width means the growth of the space of two adjacent pads.

Comparison between the results calculated by approximate formulas and FEM

The averaging coefficients of straightness errors and yaw errors calculated by the approximate formulas and FEM are shown in the Figure 9 and Figure 10, respectively. The wave number

used in the calculation is set to 0.1, 0.2, 0.3...10. We can know that all the waveforms in the two figures are similar. Furthermore, when the wave number is less than 2, the waveforms calculated by the approximate formulas and FEM are almost coincident. However, when the wave number is greater than 2, the waveforms calculated by the approximate formulas have a slight deviation from the waveforms calculated by FEM.

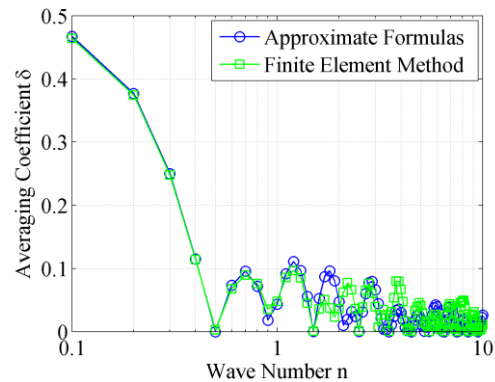


FIGURE 9. The averaging coefficient of straightness errors calculated by approximate formulas and FEM changes with the wave number, when $a=b=15\text{mm}$ and $c=4\text{mm}$.

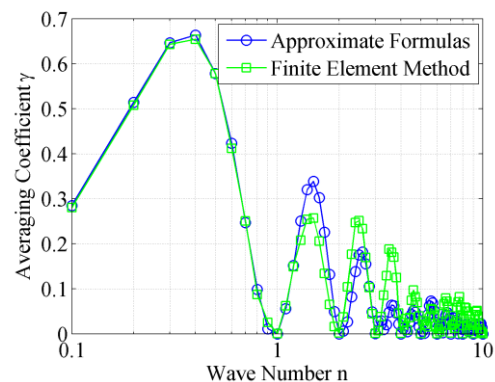


FIGURE 10. The averaging coefficient of yaw errors calculated by approximate formulas and FEM changes with the wave number, when $a=b=15\text{mm}$ and $c=4\text{mm}$.

CONCLUSIONS

Firstly, the approximate formulas for the motion errors of hydrostatic guideways under quasi-static condition were derived. Equation (13) and Equation (15) illustrate the direct relationship among structural parameters, geometric errors, and motion errors of hydrostatic guideways, reducing the difficulty of calculation.

Secondly, the discussion of the formulas was made. The overall motion errors can be obtained from the linear superposition of the parts caused

by the harmonic wave errors of the upside and downside guide rails. Besides, when the two phase angles of the harmonic waves on the upside and downside rails are reversed the motion errors can be totally or partially offset. When the number of the harmonic waves on the guide rails in the space of two adjacent pads is 0.5, 1.5, 2.5, 3.5..., the straightness errors are zero, and when the number of the harmonic waves on the guide rails in the space of two adjacent pads is 1, 2, 3, 4..., the yaw errors are zero.

Finally, the influence of land width and exhausted groove width on the error averaging effect was discussed. When the wavelength of the harmonic waves on the rails is larger than the space of two adjacent pads, the error averaging effect is weakened by the growth of land width. However, when the wavelength is smaller than the space of two adjacent pads, the error averaging effect is enhanced by the growth of land width. Moreover, the error averaging effect is weakened by the growth of exhausted groove width.

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PRECISION SURFACE MACHINING OF NEW LUNAR LASER REFLECTOR BY ION BEAM FIGURING

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INTRODUCTION

Lunar laser ranging (LLR) is the measurement of the distance between the earth and moon, which has been carried out since Apollo 11 placed a corner cube reflector (CCR) on the moon in 1969 and Apollo 14 and 15 each placed a CCR on the moon. A CCR is a reflective optical element that reflects light back toward the light source. There are two types of CCR: prismatic and hollow types. The CCRs currently placed on the moon are the prismatic type, and are arrays of prisms constructed from 100 or 300 small prisms with a diameter of 38 mm.

When a laser beam enters a prismatic CCR comprising many small prisms with an oblique incidence angle, the shape of the resulting reflected laser beam is distorted. This degrades the measurement accuracy of LLR. With this background, research on the realization of a highly accurate LLR system is being carried out, mainly by National Astronomical Observatory of Japan [1]. This LLR system involves a new CCR, which is a large single CCR with a diameter of approximately 200 mm. Moreover, the CCR is a hollow type because a prismatic CCR has

the problems of a heavy weight and non-homogeneity of the optical properties of its material. However, such a large CCR has not yet been realized.

The new CCR requires reflective surfaces with a shape accuracy of 0.1 μm . To realize the new CCR, we proposed a machining process for the CCR, as schematically shown in Figure 1 [2]. In this process, a single-crystal Si block is ground to form a hollow shape with three intersecting surfaces, and then each surface is smoothly polished [3]. Finally, the three surfaces are figured with high accuracy by ion beam figuring (IBF).

Ion beam figuring allows the machining of surfaces with high atomic-level accuracy because the removal progresses on the basis of the sputtering of atoms [4, 5]. Moreover, IBF is expected to be able to machine the corners of a CCR with high accuracy because it does not use a rigid machining tool unlike abrasive polishing. However, to our knowledge, IBF has not yet been used for the machining of three intersecting surfaces, such as the optical surfaces of the new CCR.

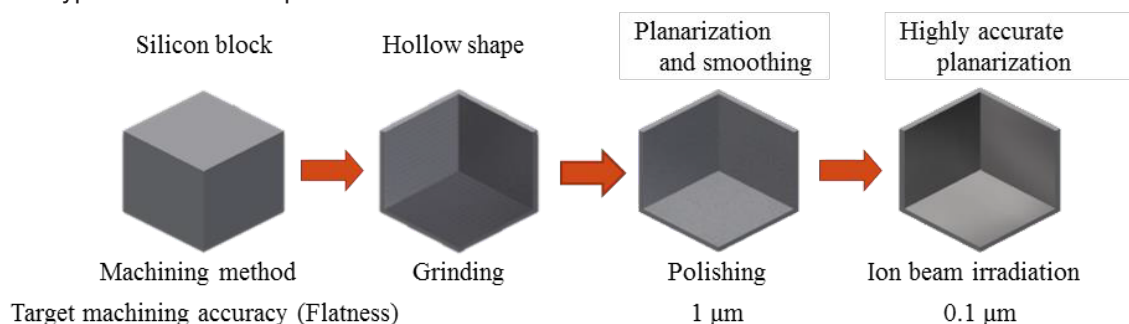


FIGURE 1. Machining process for a new corner cube reflector.

For the precision machining of three intersecting surfaces by IBF, we should consider the deposition of atoms sputtered from a surface irradiated by ion beams on other surfaces. Such deposition reduces the shape accuracy of the surfaces. Thus, we are investigating the phenomena occurring during the machining of three intersecting surfaces by IBF. On the basis of the findings obtained by the investigation, we aim to realize the new CCR with high accuracy.

In the present study, we investigate the irradiation characteristics of ion beams on two intersecting surfaces as a basic study toward the fabrication of the three intersecting surfaces of the CCR. To this end, the currents generated by ion beam irradiation in the case of two intersecting surfaces are measured during the process. Moreover, the shapes of the surfaces etched by ion beams are measured to evaluate the removal characteristics of the ion beams for the case of two intersecting surfaces.

EXPERIMENTAL APPARATUS AND METHOD

The IBF machine we used comprised an ion gun, a vacuum chamber, a four-axis stage, a shutter, a Faraday cup, a vacuum exhaust system, a mass flow controller, and a controller, as shown in Figure 2. The X and Z stages can be driven by computer numerical control, which allows us to control the moving distance, moving position, and residual time of the ion beams. The Y and θ stages are manual stages, which can be used to change the irradiation distance between the ion gun and the workpiece surface and the angle of the workpiece relative to the ion beams. On the θ stage, two workpieces can be placed at right angles to each other.

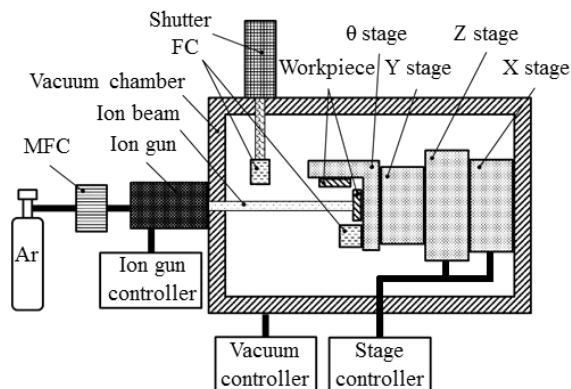


FIGURE 2. Schematic of ion beam figuring machine used in study.

In processing experiments, ion beams were irradiated to two intersecting plates with an accelerating voltage of 4 kV. The two intersecting plates were single-crystal silicon wafers with a size of 50 mm $\times$ 50 mm $\times$ 0.7 mm, which were set at right angles to each other. The plate irradiated with ion beams is referred to as the target plate, and the plate set perpendicular to the target plate is referred to as the perpendicular plate, as shown in Figure 3. The profile of the ion beams was measured with a Faraday cup mounted on the stage, which showed that the diameter of the ion beams was 11 mm.

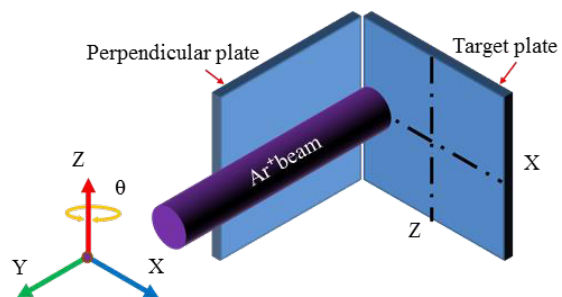


FIGURE 3. Experimental arrangement of two plates processed in this study.

In the present study, the ion beams were irradiated perpendicular to the target surface, although the process can be performed with various incidence angles to the target surface using the IBF machine. An advantage of the irradiation of perpendicular ion beams is that it will be easy to operate the IBF machine in the actual production of the CCR.

MEASUREMENT OF CURRENTS FLOWING THROUGH THE PLATES DURING ION BEAM IRRADIATION

Current measurement for intersecting plates

To investigate the effect of ion beam irradiation on the two intersecting surfaces, the currents flowing through the two plates were measured during the ion beam process. Figure 4 shows a schematic of the experimental apparatus used for current measurement. As shown in Figure 4, the workpieces were connected with wires to the ground through resistors. In this arrangement, the voltage applied to the resistors was measured during ion beam irradiation. The current was calculated by dividing the measured voltage by the resistance value. The ion currents flowing through the perpendicular and target

plates were measured for various irradiation positions of the ion beams, which were varied by moving the two intersecting plates using the X stage.

Figure 5 shows the experimental results. It is found that the current flowing through the target plate tends to decrease when the ion beams approach the perpendicular plate. As shown in Figure 5, the decrease in current begins from $X = 10$ mm. The reason for the decrease in current is probably that the ion beams gradually move out of the target plate as they approach the perpendicular plate.

Considering the diameter of the ion beams of 11 mm, the decrease in the current must begin at $X = 5.5$ mm, in contrast to the experimental result. Therefore, the reason for the decrease in current from $X = 10$ mm cannot be explained only by the ion beams protruding from the target plate. This will be discussed again after the next section.

As shown in Figure 5, the current flowing through the perpendicular plate increases to a peak at $X = 4$ mm. To clarify the reason for this change in the current through the perpendicular plate, we performed the experiments described in the following section.

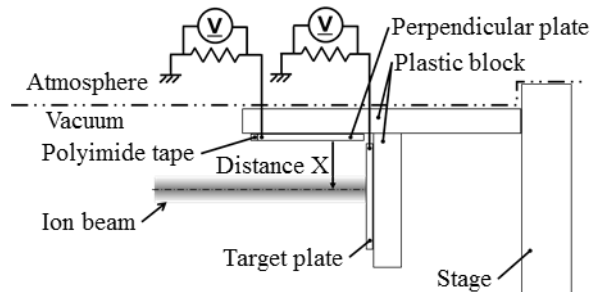


FIGURE 4. Experimental apparatus used for current measurement of two intersecting plates.

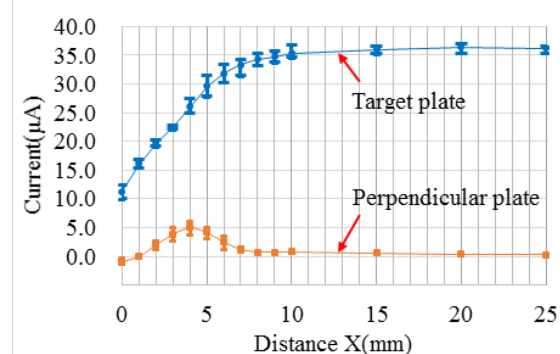


FIGURE 5. Experimental results of current measurement for two intersecting plates.

Current measurement of perpendicular plate

The following are two possible reasons for the increase in the current through the perpendicular plate shown in Figure 5. The first is the effect of recoil particles from the irradiated surface. The second is the incidence of ion beams passing in front of the perpendicular plate. To investigate the likelihood of the two reasons, we performed an experiment, in which the current through the perpendicular plate was measured while the ion beams were passed in front of the perpendicular plate without the target plate, as shown in Figure 6. The target plate was removed to prevent the effect of the recoil particles from the target surface.

Figure 7 shows the experimental results. As shown in Figure 7, the current is detected even when the recoil particles are not generated from the target surface in the absence of the target plate. Therefore, the current through the perpendicular plate is considered to be due to the incidence of the ion beams passing in front of its surface. Moreover, it is found that the current through the perpendicular plate in the presence of the target plate is larger than that without it. This indicates that the perpendicular plate is also affected by the recoil particles from the target plate.

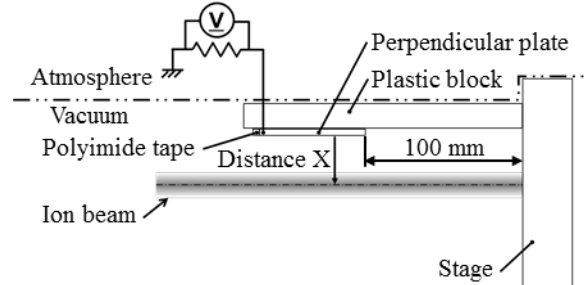


FIGURE 6. Experimental apparatus used for current measurement of perpendicular plate.

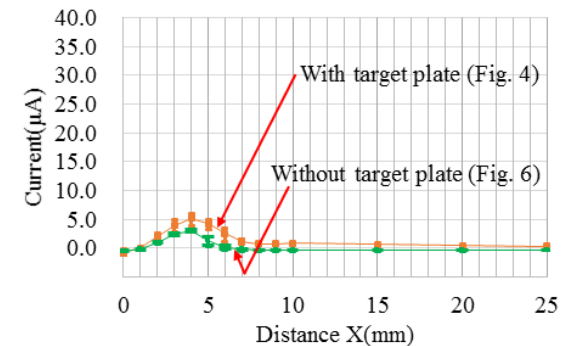


FIGURE 7. Experimental results of current measurement for perpendicular plate with and without target plate.

ETCHING OF TWO INTERSECTING PLATES BY ION BEAMS

Experimental purpose and method

In the previous section, we showed that the current passing through the target plate decreases when the ion beams approach the perpendicular plate. Next, to investigate the effect of the perpendicular plate on the etching profile of the target surface, we etched two intersecting plates with ion beams. The processing conditions are shown in Table 1.

TABLE 1. Processing conditions.

Processing time (min)	180
Incidence angle (deg)	0
Working distance (mm)	200
Distance X (mm)	3-6

TABLE 2. Measurement conditions.

X-axis measurement pitch (μm)	100
Z-axis measurement pitch (μm)	100
Laser spot size (μm)	0.4

The surface profiles of the plates after the process were measured by a three-dimensional measuring instrument employing a laser probe (Mitaka Kohki Co., Ltd., NH-3) [6]. The measurement conditions are shown in Table 2.

Results and discussion

Examples of the measured profiles of etched surfaces are shown in Figure 8. In Figure 8, the center points of the etched areas are indicated by arrows, which were determined as the minimum of the sixth-order curve used to approximate the etched profile. As shown in Figure 8, the center point of each etched area is found to be shifted toward the perpendicular plate from the target point of the ion beams; the target point was the center of each ion beam, which was determined using the Faraday cup when there was no perpendicular plate. This finding indicates that the ion beams are attracted to the perpendicular plate. Therefore, the decrease in the current through the target plate from $X = 10$ mm shown in Figure 5 is considered to be due to the attraction of the ion beams to the perpendicular plate.

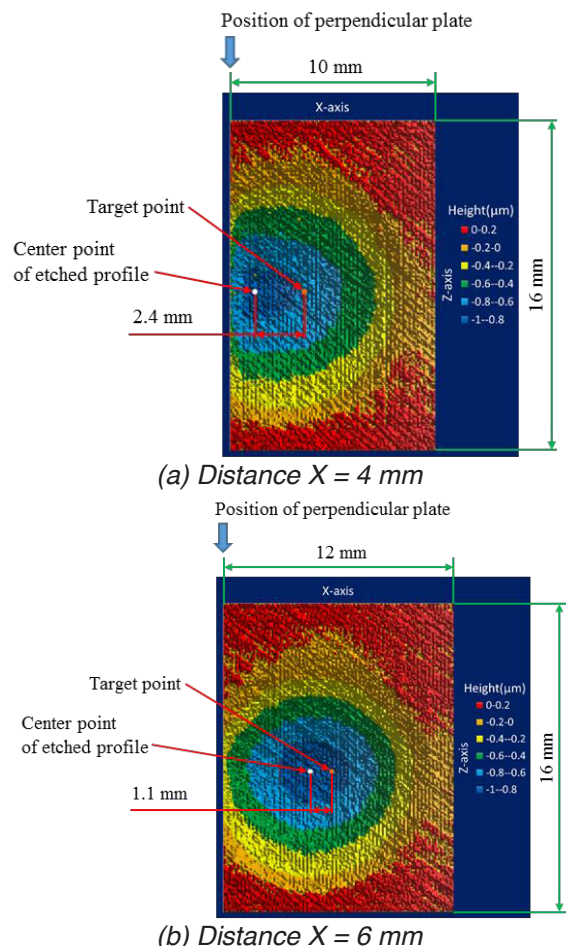


FIGURE 8. Experimental results for etching of two intersecting plates. Profiles of target plates are shown.

CONCLUSION

A fundamental study was conducted toward the realization of a new corner cube reflector (CCR) for lunar laser ranging. The application of ion beam figuring (IBF) to the fabrication process of the CCR was examined by investigating the irradiation of ion beams to two intersecting plates. The shapes of the surfaces processed by the ion beams were measured to examine the removal characteristics of the ion beams for the case of two intersecting plates. As a result, the center point of the etched area was found to be shifted toward the perpendicular plates from the target point of each ion beam. This finding indicates that the ion beams are attracted to the perpendicular plate. Considering the findings obtained in this study, it is expected that the CCR can be fabricated with high accuracy by IBF under compensation of the shift of the irradiation positions of the ion beams.

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MULTISTAGE SUPERFINISHING OF SAPPHIRE WITH VITRIFIED-BONDED DIAMOND SUPERABRASIVE STONES

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ABSTRACT

Sapphire is widely used for the LED substrate. In the planarization process of sapphire, lapping is used as the former process of polishing. However, the machining time of lapping takes about 30 minutes because sapphire is hard material. The superfinishing process can be substitute for the lapping. Superfinishing is a fixed abrasive machining, and it is possible to process sapphire by short time.

The target surface roughness of the lapping process is 2nmRa. Therefore, superabrasive stones are used for the superfinishing. Then, vitrified-bonded diamond superabrasive stones with different grain diameter and different grade were developed, and superfinishing performances of those stones were compared. As a result, the diamond superabrasive stones with lower grades are more suitable stones as the former process of the polishing. However, those stones reached the stone life when the superfinishing with were done from the ground surface that was the former process of the lapping, and surface roughness were not able to be achieved. And some deep scratches of the former process remained when the amount of superfinishing process was not enough.

At that point, multistage superfinishing was examined by combining some diamond superabrasive stones with different grain diameter. Finally, the removal amount estimation method was developed based on the real contact pressure presumption under the multistage process aiming at the achievement of the target roughness by short time.

MATERIALS AND METHODS

(1) Superfinishing performance of diamond superabrasive stones with different grain diameter and different grade

Diamond superabrasive stones with different grain diameter and different grade were

developed. TABLE 1 shows the superfinishing conditions and specification of the stones.

The superfinishing performance of those stones were examined. The pre-finished surface was made as ground surface that was the former process of the lapping. The stone speed of the superfinishing was constant, and the stone pressure was varied.

TABLE 1. Stone specification and superfinishing conditions.

Stone specification	SD 2000 I 110 V
	SD 4000 I 110 V
	SD 6000 I 110 V
	SD 8000 I 110 V
	SD 12000 L 105 V
	SD 20000 L 105 V
Superfinishing pressure	0.11-0.90 MPa
Stone speed	452 m/min
Workpiece speed	77 m/min
Oscillation	16.7 Hz
Amplitude	0.5 mm
Superfinishing time	60 s
Superfinishing fluid	Water (rust inhibitor 1%)
Workpiece	Sapphire

(2) Multistage superfinishing of sapphire with vitrified-bonded diamond superabrasive stones

Each stone reached the stone life in (1), and the target surface roughness was not able to be achieved. And some deep scratches of grinding process remained when the amount of the superfinishing process was not enough. Therefore, the roughness of former process is important. Then, some diamond superabrasive stones with different grain diameter were combined, and the multistage superfinishing was examined. Table 2 shows the process of the multistage superfinishing. The stone pressure was the highest pressure that was able to be used in (1).

TABLE 2. Multistage superfinishing processes.

	Stage 1	Stage 2	Stage 3	Stage 4
A	SD 4000	SD 20000		
B	SD 2000	SD 4000	SD 20000	
C	SD 2000	SD 6000	SD 20000	
D	SD 2000	SD 4000	SD 6000	SD 20000

(3) Development of finish amount estimation method based on real contact pressure presumption under process

Stones combinations in (2) were chosen in heuristic method, and it cannot be the optimal process. It is necessary to examine a lot of combinations to achieve the target roughness in an optimal way. However, it is difficult to examine it experimentally because the number of combination is huge. Then, the removal amount estimation method based on the real contact pressure presumption under the multistage process is developed based on the experimental result obtained in (2). It is necessary to presume the change in stone working surface area during a process and the change in the rate of the contact area of the sapphire to presume the real contact pressure under the process. The stone working surface area under a process is defined as the elastic deformation of stone shape by the stone pressure[1]. Sapphire is much harder than stones and can be regarded as rigid body. At the beginning of superfinishing, the highest point of the stone touch with sapphire surface, which can be regarded as flat. The highest point of the stone deforms elastically by the stone pressure. Then, other high points of the stone contact with sapphire and deform one after another. Those deformations of the stone have the strength of elastic force. The increase of the stone deformation brings the increase of contact area between the stone and sapphire surface. Because of this, as the stone deformation increases, total amount of elastic force increases. The stone can be deformed until the elastic force becomes equal to the force of stone pressure. The area at this moment is the real working surface area of the stone. The working surface area ratio is calculated by the area. The equation to calculate the amount of elastic deformation is as follows:

$$A \cdot P_s = \sum_{h=0}^{h_e} a \cdot f(h) \cdot E \cdot \left(\frac{h_e - h}{l - h} \right)$$

where A is the area of measurement, a is the area of each pixel, f(h) is the number of pixels whose depth is h, E is the elastic modulus of the stone, and l is asperity of the stone.

FIGURE 1 shows the contact model between stone and sapphire. The real contact area between stone and sapphire is affected by the surface roughness of sapphire. The rate of the contact area of the sapphire under a process is decided by the amount of removal till the process, the depth of cut of the stone to the sapphire, and the roughness of former process. Therefore, the amount of removal increases and the real contact pressure decreases.

The depth of cut of stone to the sapphire was investigated by using sapphire surface of 2nmRa. FIGURE 2 shows the dependence of depth of cut as a function of grain diameter. The depth of cut of stone to the sapphire can be exponentially approximated by the grain diameter of the stone. The minimum surface roughness of the stone can be calculated by the depth of the cut of the stone.

Consequently, the amount for a given short time can be presumed by presuming the real contact pressure for a given short time.

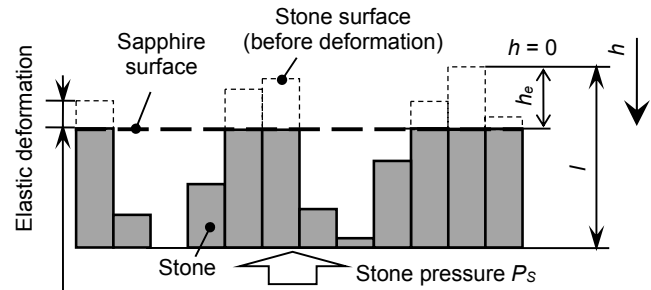


FIGURE 1. Contact model between stone and sapphire.

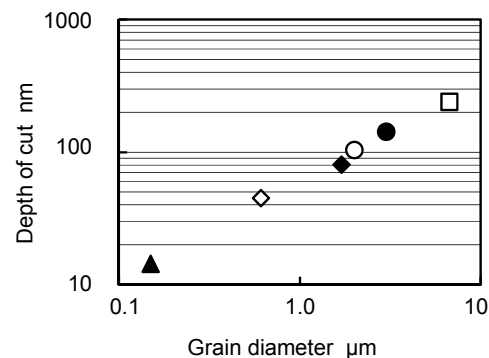


FIGURE 2. Dependence of depth of cut as a function of grain diameter.

RESULTS

(1) FIGURE 3 shows the comparison of surface roughness of sapphire in superfinishing with vitrified-bonded diamond superabrasive stones. Each stone reached the stone life, and the target surface roughness was not able to be achieved. And scratches of the former process remained when the amount of superfinishing process was not enough. Therefore, the amount of a process must be larger than the depth of scratches of the former process.

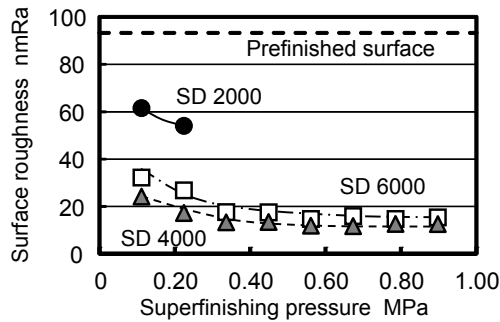


FIGURE 3. Comparison of surface roughness of sapphire in superfinishing with each stone.

(2) FIGURE 4 shows superfinished surface of sapphire in each process of multistage superfinishing. Scratches of pre-finished surface remained in stage 1 of process A (SD4000 stone). Thus SD2000 stone (stage 2 of process A) was unable to remove the scratches because the amount of superfinishing with SD20000 is very small. Therefore, SD2000 stone was used before SD4000 in stage 1 of process B and D because the amount of superfinishing with SD2000 is much larger than that of SD4000. SD2000 stone removed the scratches of pre-finished surface. Though the scratches after SD2000 and SD4000 stones were relatively small, SD20000 stone was not able to remove them in stage 3 of process B. Then, SD6000 stone was substitute for SD4000 stone in stage 2 of process C. After stage 3 of process C with SD20000 stone, shallow scratches remained on sapphire surface. Finally, SD6000 stone was used after SD4000 stone and before SD20000 stone in process D. In process D, SD6000 stone was able to remove all the scratches of SD2000 and SD40000 stones. After that SD20000 stone was able to remove the scratches of SD6000 stone. Target surface roughness was achieved in process D.

The target roughness 2nmRa was achieved in 8 minutes of multistage superfinishing. FIGURE 5

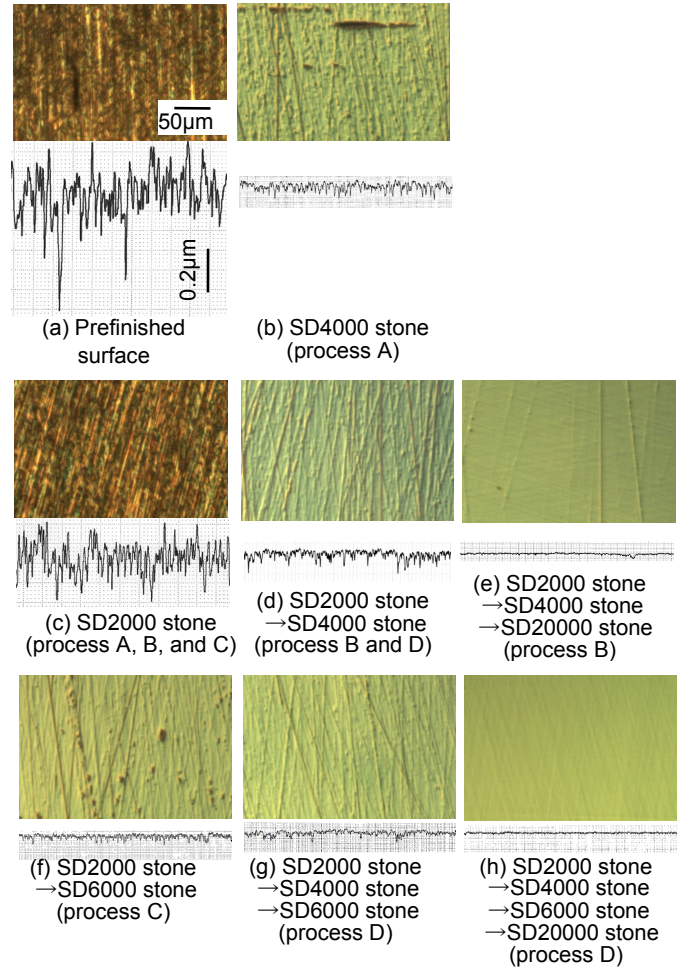


FIGURE 4. Superfinished surface of sapphire in each process of multistage superfinishing.

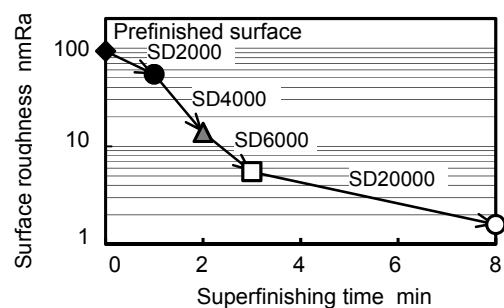


FIGURE 5. Improvement of surface roughness in multistage superfinishing. (process D)

shows the improvement of surface roughness in multistage superfinishing of process D. The minimum surface roughness of SD20000 stone can be calculated as 2nmRa by the fact that the depth of the cut of the stone is 10nm.

(3) Then, we evaluated the calculated real contact pressure and estimated the amount of sapphire removal. As mentioned in (1), the amount for a given short time can be presumed by presuming the real contact pressure for a given short time. The amount during the process is necessary to calculate the real contact pressure. Hence iterative calculation method was used to presume the amount in each minute time. The amount was presumed by presuming the area of stone working surface and the rate of the contact are of sapphire during superfinishing with each stone.

FIGURE 6 shows a laser microscopic image of SD4000 stone before superfinishing. FIGURE 7 shows the calculated stone working surface area of SD4000 stone at the beginning of superfinishing. Calculated depth of elastic deformation was $2.2\mu\text{m}$, and calculated working surface area ratio was 1.8%.

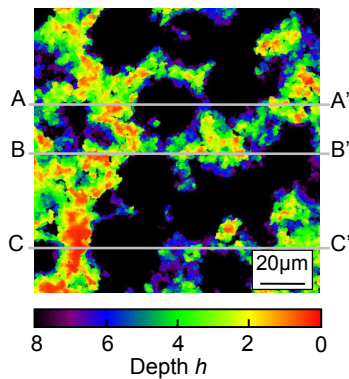


FIGURE 6. Laser microscopic image of SD4000 stone before superfinishing.

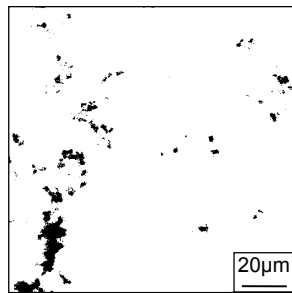


FIGURE 7. Calculated stone working surface area of SD4000 stone at the beginning of superfinishing. ($h_e=2.2\mu\text{m}$, working surface area ratio: 1.8%)

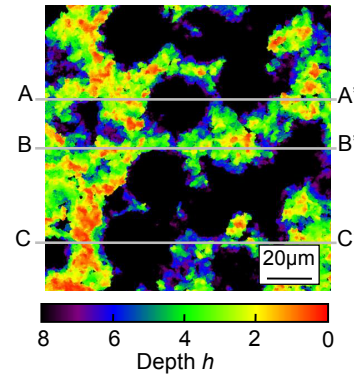


FIGURE 8. Laser microscopic image of SD4000 stone after superfinishing.

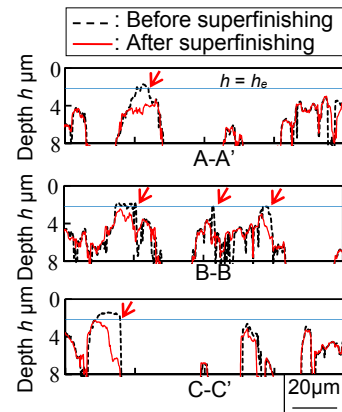


FIGURE 9. Comparison of SD4000 stone profiles before and after superfinishing.

FIGURE 8 shows the laser microscopic image of SD4000 stone after superfinishing. FIGURE 9 shows the comparison of SD4000 stone profiles before and after superfinishing. The change of the profiles confirms the real stone working area based on calculation of elastic deformation of stone surface.

Calculated amount of each stone was corresponded to measured amount by the method.

CONCLUSIONS

Results indicate that the change of stone working surface area under superfinishing is defined as the elastic deformation of stone shape by the stone pressure and the change of rate of the contact area of the sapphire under superfinishing is decided by combining the amount of removal till the process and the depth of cut of the stone.

Subsequently we conducted more efficient multistage superfinishing of sapphire.

ACKNOWLEDGEMENT

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STUDY ON 3D PROFILE-CONTROLLED PATTERNING IN CYLINDRICAL BLASTING

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INTRODUCTION

Abrasive blasting is mainly used for deburring and improvement of surface property. On the other hand, the method can be micromachining under proper machining conditions [1]. In this paper, dynamic pressure grooves on the shaft of a fluid dynamic pressure bearing was manufactured by cylindrical blasting.

Figure 1 shows the structure of a fluid dynamic pressure bearing. The bearing is employed for spindle motors of hard disk drive for its high-accuracy revolutions. There are micro herring-bone grooves on the shaft for generating dynamic pressure in the lubricant fluid. In regards to the dynamic pressure grooves, there is a research about thrust dynamic pressure bearing with varying depth grooves [2]. The groove depth decreases from the ends of the groove to its central part as shown in Figure 2. The groove profile is expected to gather lubricant fluid into its central part efficiently and increase the dynamic pressure.

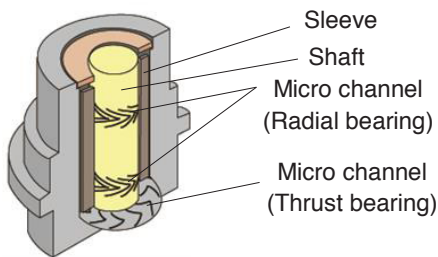


FIGURE 1. Structure of a fluid dynamic pressure bearing.

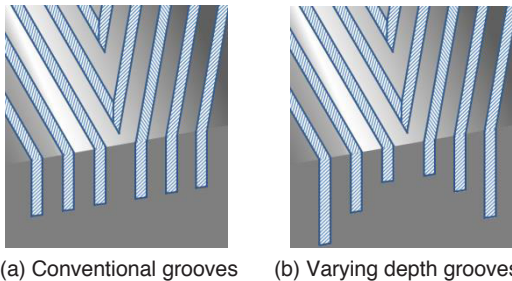


FIGURE 2. 3D profiles of herring-bone grooves.

This advantageous effect is estimated to act on radial bearings with varying depth grooves too.

These grooves are mainly manufactured by chemical etching, electrochemical machining, cutting and form rolling [3]. However, chemical etching and electrochemical machining are difficult to control 3D machining profiles with high accuracy. Cutting and form rolling have problems in tool lives and manufacturing costs. Abrasive blasting can replace the conventional methods, because it does not have these problems.

The purpose of this study is to control machining profiles in cylindrical blasting and to simulate the profiles. Then, the varying depth grooves were manufactured by cylindrical blasting as examples of profile-controlled patterning.

EXPERIMENTAL SETUP AND CONDITIONS

Figure 3 shows the experimental setup of cylindrical blasting. The removal machining was performed by blasting abrasive grains with compressed air. The nozzle was fed in the direction of x-axis at the same time. In regards to this x-axis direction feed, control of 3D machining profiles was enabled by changing the nozzle feed speed continuously. The stock removal is inversely proportional to the nozzle feed speed. Then, the simulation of machining profiles in cylindrical blasting was invented.

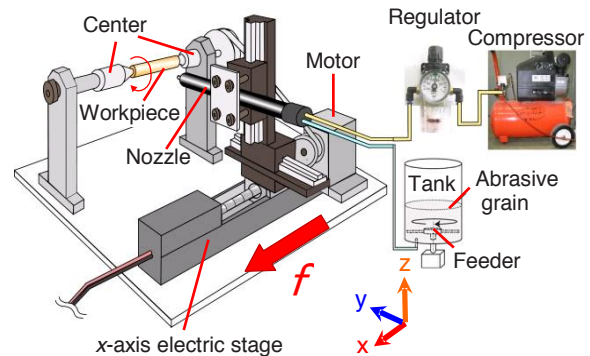


FIGURE 3. Experimental Setup.

TABLE 1. Experimental conditions.

Abrasive grain	GC 1000
Workpiece	C45 $f 10 \times 45$ mm
Blasting pressure	$P = 0.5$ MPa
Blasting stand-off distance	$y = 5$ mm
Inside diameter of nozzle	$d_n = 2$ mm
Peripheral speed of workpiece	$V_w = 18.8$ m/min
Mask	Vinyl chloride (thickness : 75 μ m)

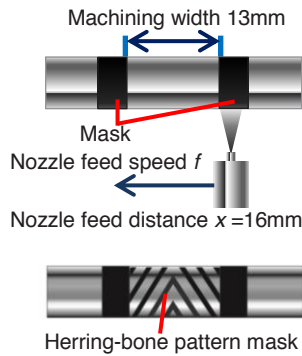


FIGURE 4. Workpieces with and without herring-bone pattern mask.

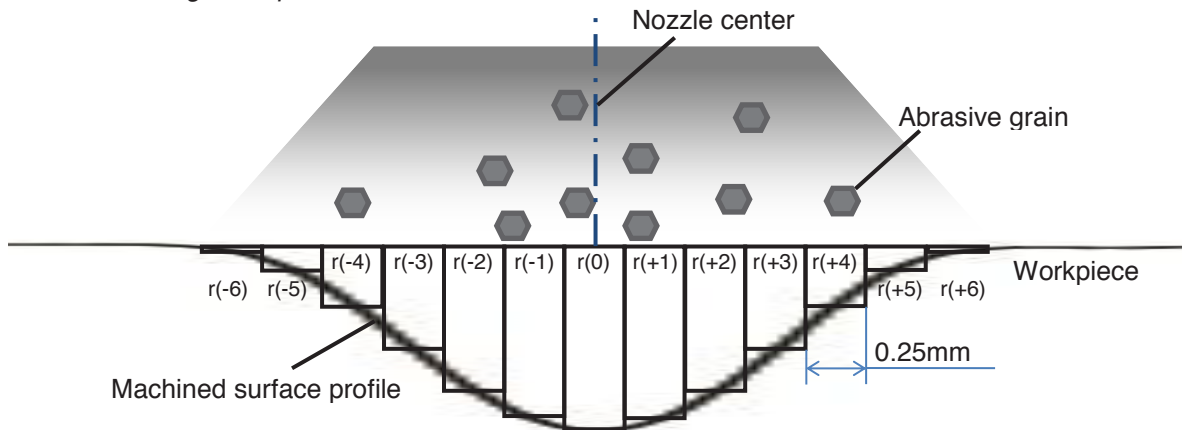


FIGURE 5. Division of blasting area.

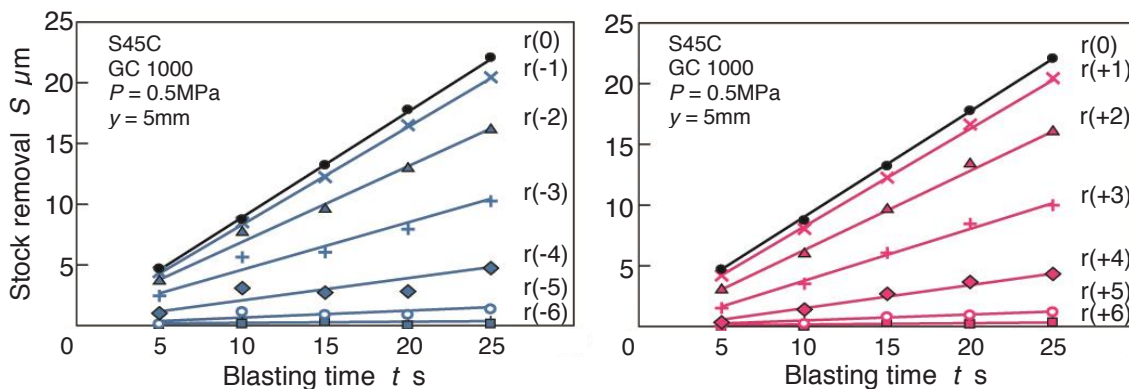


FIGURE 6. Relationship between stock removal and blasting time.

Table 1 shows the experimental conditions. Two kinds of Workpieces which were machined in this experiment are shown in Figure 4. The workpiece without herring-bone pattern mask was used to examine the removal machining speed in cylindrical blasting and to verify the accuracy of the simulation. The workpiece with herring-bone pattern mask was used for patterning the grooves. The nozzle feed distance was set 16 mm. The actual machining width was 13mm.

SIMULATION METHOD AND EXPERIMENTAL RESULT

First, removal machining speed of blasting in this experiment was examined by cylindrical blasting with no nozzle feed. The blasting area was divided into thirteen areas. The width of each area is 0.25mm. Figure 5 schematically shows the division of the blasting area. Each area was named as shown in it. The result of this examination is shown in Figure 6.

TABLE 2. Stock removals of each area.

Area	Stock removal μm	Area	Stock removal μm
$r(0)$	$S_{r(0)} = 0.876 t$	$r(0)$	$S_{r(0)} = 0.876 t$
$r(-1)$	$S_{r(-1)} = 0.802 t$	$r(+1)$	$S_{r(+1)} = 0.822 t$
$r(-2)$	$S_{r(-2)} = 0.603 t$	$r(+2)$	$S_{r(+2)} = 0.669 t$
$r(-3)$	$S_{r(-3)} = 0.358 t$	$r(+3)$	$S_{r(+3)} = 0.437 t$
$r(-4)$	$S_{r(-4)} = 0.143 t$	$r(+4)$	$S_{r(+4)} = 0.205 t$
$r(-5)$	$S_{r(-5)} = 0.045 t$	$r(+5)$	$S_{r(+5)} = 0.061 t$
$r(-6)$	$S_{r(-6)} = 0.003 t$	$r(+6)$	$S_{r(+6)} = 0.017 t$

Blasting time t s

Since each stock removal is in proportion to the blasting time approximately, the slopes of these straight lines were defined as stock removal per a second at any blasting areas. Table 2 shows stock removals of each area obtained from the data of Figure 6.

When the nozzle feed speed set at 0.25mm/s, the nozzle passes an area in a second. Then, relationship equations between nozzle feed speed and stock removal are able to establish. If the nozzle feed speed is doubled, then the blasting time is halved. Therefore, the stock removal is halved. The nozzle feed speed f is inversely proportional to the stock removal. The relationship between nozzle feed speed and stock removal was formularized by following equation.

$$S_n = \frac{a_n}{f} \quad (a_n \text{ is constant}) \quad (1)$$

This a_n was calculated by multiplying 0.25 and the coefficients of Table 2 together.

ARC PROFILE MACHINING

The nozzle feed distance was defined as x mm ($0 \leq x \leq 16$). Figure 7 shows the relationship between nozzle feed speed f and blasting time t . In this experiment, maximum feed speed f_1 mm/s, minimum feed speed 0 mm/s, acceleration time t_1 s, deceleration time $t_2 - t_1$ s, and the total nozzle feed distance 16 mm were set. The motion was carried out with the constant acceleration and deceleration. The hatching area of Figure 7 represents the nozzle feed distance x . The following equations show relationship between nozzle feed distance x and nozzle feed speed f .

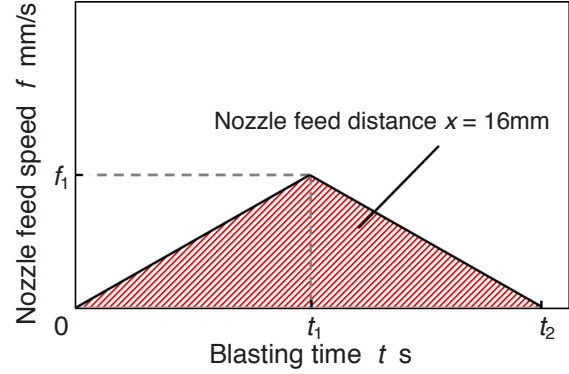


FIGURE 7. Relationship between nozzle feed speed and blasting time.

$$x = \frac{1}{2} f t \quad [x \leq 8\text{mm}] \quad (2)$$

$$x = 16 - \frac{1}{2} f (80 - t) \quad [8\text{mm} \leq x \leq 16\text{mm}] \quad (3)$$

The following equations show relationship between blasting time t and nozzle feed speed f .

$$t = \frac{t_1}{f_1} f \quad [x \leq 8\text{mm}] \quad (4)$$

$$t = t_2 - \frac{t_1}{f_1} f \quad [8\text{mm} \leq x \leq 16\text{mm}] \quad (5)$$

The following equations are obtained by referring to Eq. (2) ~ (5)

$$x = \frac{t_1}{2 f_1} f^2 \quad [x \leq 8\text{mm}] \quad (6)$$

$$x = 16 - \frac{t_1}{2 f_1} f^2 \quad [8\text{mm} \leq x \leq 16\text{mm}] \quad (7)$$

The following equations are obtained by referring to Eq. (1), Eq. (6), and Eq. (7).

$$S_n = a_n \sqrt{\frac{t_1}{2 f_1 x}} \quad [x \leq 8\text{mm}] \quad (8)$$

$$S_n = a_n \sqrt{\frac{t_1}{2 f_1 (16 - x)}} \quad [8\text{mm} \leq x \leq 16\text{mm}] \quad (9)$$

Eq. (8) and Eq. (9) show simulated machining profiles.

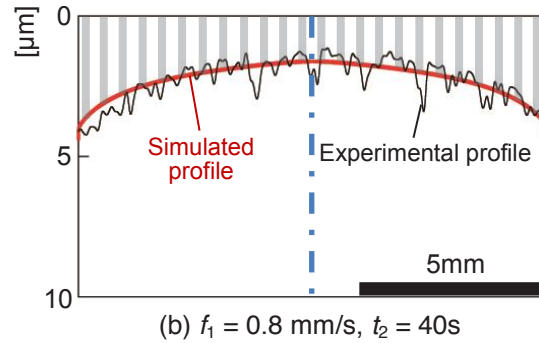
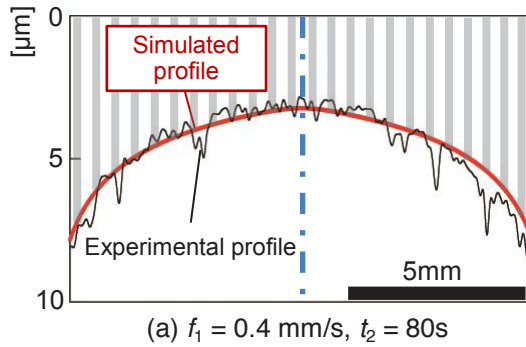


FIGURE 8. Experimental profiles and simulated ones (arc profile).

EXPERIMENTAL RESULT

Above-mentioned simulation was performed on two kinds of experimental conditions. The simulated machining profiles were compared with the experimental ones under the same experimental condition. Figure 8 shows the simulated arc profiles and the experimental ones. The simulated profiles agreed well with the experimental ones.

PATTERNING OF HERRING-BONE GROOVES

Finally, the herring-bone grooves patterning was carried out by blasting with mask. However, the stock removal decreased as compared to the simulation. This fact suggests that the mask affected the machining progression. Then, the machining profiles considering the influence of mask were simulated. We found that the stock removal decreases about 11% for the effect of the mask by fundamental experiments. Figure 9 shows the simulated profiles of groove bottom and the experimental herring-bone groove profiles using the mask. The experimental conditions are the same with arc profile machining. The simulated profiles agreed well with the experimental groove bottom profiles. These results suggest that blasting is able to control machining profiles in patterning.

CONCLUSIONS

The purpose of this study is to control machining profiles in cylindrical blasting and to simulate the profiles. The simulated profiles agreed well with the machining profiles under any conditions. 3D profile-controlled machining in cylindrical blasting was enabled by controlling the nozzle feed speed.

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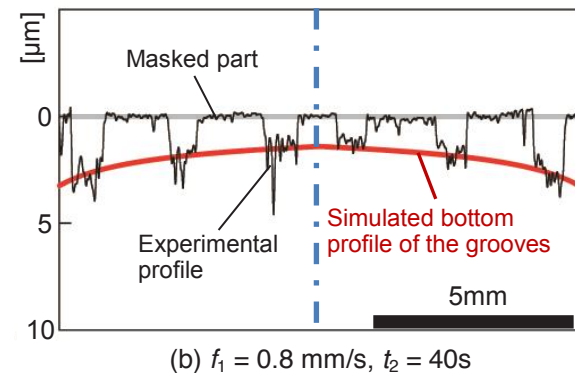
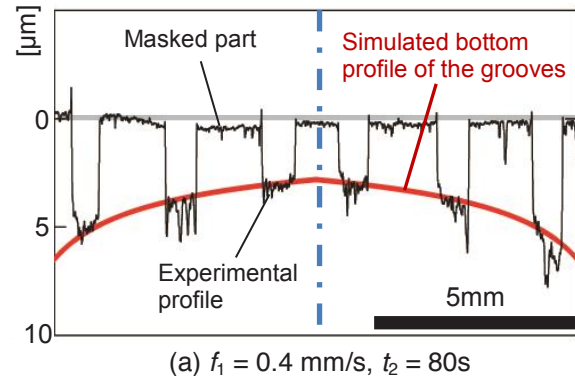


FIGURE 9. Experimental profiles and simulated ones (herring-bone grooves).

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TURNING-LASER HYBRID MACHINING OF MICRO-PATTERNS ON A ROLL MOLD

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INTRODUCTION

Recently, there has been growing interest and demand for functional thin films with precise micro-patterns in large area for application of flat panel display, printed electronics and decoration. This has allowed the roll-to-roll forming process to be widely used in fabrication of the large area thin films. The large diameter roll dies with precise micro-patterns can be usually machined using ultra-precision roll die lathes. The most conventional roll lathes conduct only one machining process such as turning using a diamond tool or laser machining. Diamond turning process is capable of generating fine finished patterns, but only continuous patterns such as V-shaped or square patterns are common. Laser machining process can create various patterns, but the machined surface is poor.

To cope with increasing demand for more complicated non-continuous micro-patterns for the applications in optical films, flexible circuit and in-mold decoration films, a new hybrid roll die lathe which can conduct laser machining, fast tool servo (FTS) cutting and diamond turning was developed by authors [1, 2].

In this research, diamond turning and laser hybrid machining based micro-patterning process on a roll die for optical films using the developed precise hybrid roll die lathe is introduced. Actually formed film by the micro-patterns which are machined using the developed turning-laser hybrid roll lathe is presented to evaluate the introduced process.

HYBRID ROLL LATHE

Figure 1 and figure 2 show the conceptual design of the developed hybrid roll lathe and the actually developed hybrid roll lathe. Turning tool and FTS machining modules are located on the B axis stage at the front side of the roll, and the laser machining module is located on the X2 axis stage at the rear side of the roll. Z axis stage travels along the length of roll and carries X1 and X2 axis stages. The spindle in the headstock that rotates the roll and Z axis stage

are supported by oil-hydrostatic bearings to improve accuracy of the machine. Maximum size of a roll which can be machined is diameter of 500mm and length of 2000mm. Maximum roll rotating speed is 300RPM.

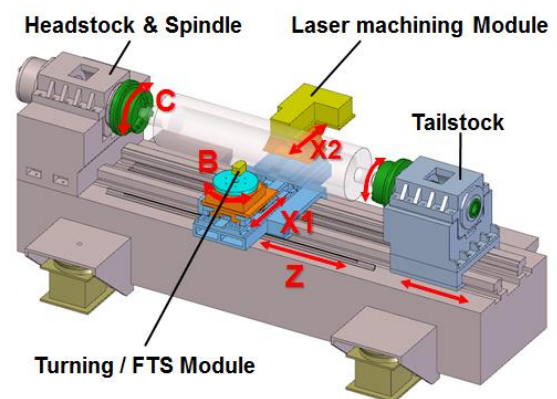
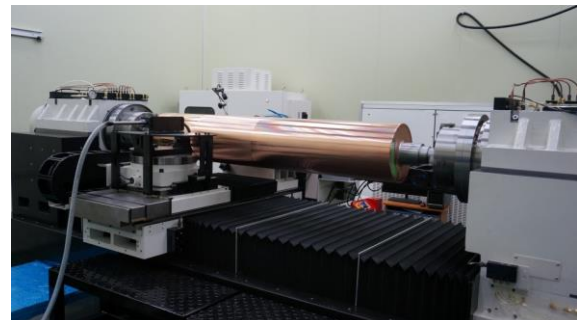


FIGURE 1. Design of the developed hybrid roll lathe



(a) Front side



(b) Rear side

FIGURE 2. Developed hybrid roll lathe

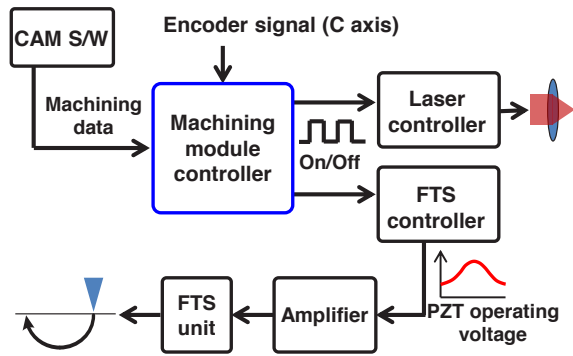


FIGURE 3. Schematic diagram of the developed machining module control system

The axes as shown in figure 1 and turning module of the roll lathe are controlled by the main motion controller (Power PMAC, Delta Tau). In order to create precise non-continuous micro-patterns using laser or FTS machining module at the desired position on the roll surface where indicated by machining data, the additional control system for machining modules was developed. Figure 3 shows the schematic diagram of the developed laser and FTS machining module control system. Binary machining data is generated by the developed CAM software which imports images or CAD models. The machining module controller generates a trigger signal for operating the laser or FTS units at the precise moment when the patterning is required by considering real-time encoder signals of the rotational axis and the machining data. The shape of each machined pattern is controlled by the laser controller or FTS controller.

HYBRID MACHINING OF MICRO-PATTERNS

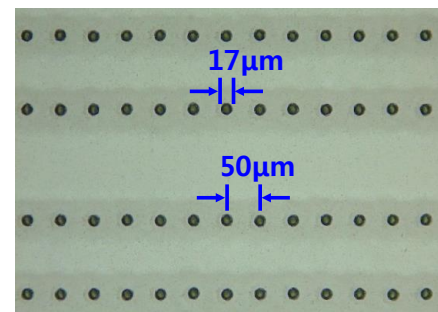
In order to evaluate the developed hybrid machining system an experimental machining of micro-patterns for optical films was conducted. Target patterns to be machined, which aim to prevent a wet-out phenomenon caused by the surface-contact, are micro-dimples on peaks of V-shaped grooves with pitch of $50\mu\text{m}$. Conventional method for machining of the target patterns requires both a turning roll lathe and a laser roll lathe. This causes alignment error between the grooves and dimples. In this research, micro-dimples were machined by laser and then V-shaped grooves were machined between the dimples using diamond turning tool in the developed hybrid roll lathe.

Copper coated roll with diameter of 320mm and length of 2m was used to the experiment. The roll surface was machined for mirror-like

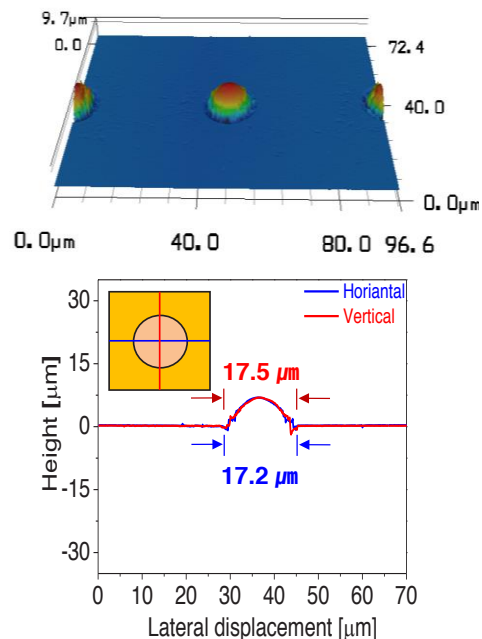
finishing using turning process before laser machining.

Ytterbium fiber laser, which has an output wavelength of 1064nm, pulse duration of 60ns and output power of 100 W, was used for laser machining of the micro-dimples.

Figure 4(a) shows the generated micro-dimple arrays on the roll surface, and figure 4(b) shows an image of single dimple of ultraviolet-curing resin formed by the machined roll die and its cross-sectional profile measured by a confocal microscope. Figure 5(a) shows the fabricated micro-patterns on the roll surface after V-shape grooving between the dimples and the magnified image. And figure 5(b) shows an actually formed ultraviolet-curing resin by the roll die and its measured cross-sectional profile. It is shown that V-shaped grooves with dimples at their peaks are precisely fabricated on the roll.

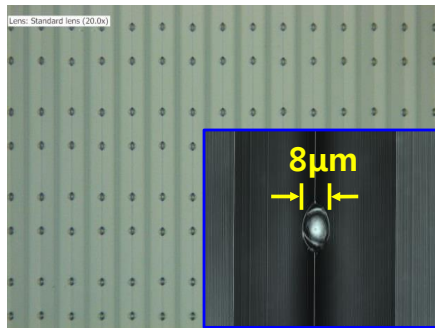


(a) Machined micro-dimple arrays on the roll die

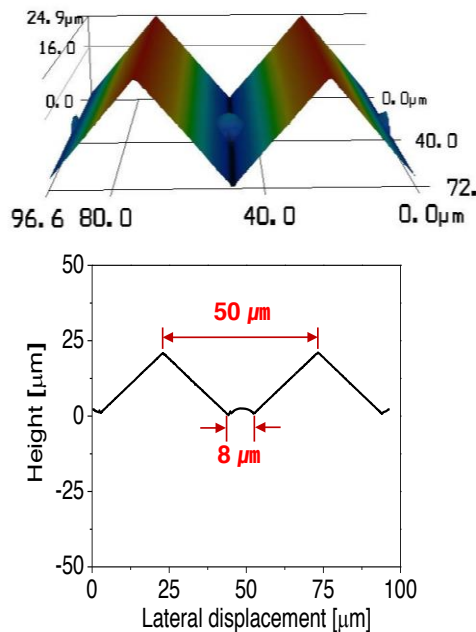


(b) Formed ultraviolet-curing resin

FIGURE 4. Micro-dimple arrays created by laser machining



(a) Machined hybrid patterns on the roll die



(b) Formed ultraviolet-curing resin

FIGURE 5. Fabricated hybrid micro-patterns

CONCLUSIONS

In this research, the turning-laser hybrid machining process for precise micro-patterning on a roll mold using the newly developed hybrid roll die lathe is introduced. Experimental machining of hybrid micro-patterns for functional optical films was conducted. The machining results demonstrate that the proposed process using the developed hybrid roll lathe can be successfully applied to patterning of the more complicated non-continuous micro-patterns, which cannot be easily obtained from using conventional roll lathes based on only process such as a turning or laser machining process.

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PERIOD-N BIFURCATIONS IN MILLING: NUMERICAL AND EXPERIMENTAL VERIFICATION

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INTRODUCTION

Many years of machining simulation and measurement research have led to a comprehensive understanding of milling process dynamics. As early as 1946 Arnold studied chatter in steel machining [1]. Doi and Kato described self-excited vibrations using time-delay differential equations in 1956 [2]. During this time, the notion of “regeneration of waviness” was promoted as the feedback mechanism (time-delay term), where the previously cut surface combined with the instantaneous vibration state dictates the current chip thickness, force level, and corresponding vibration response [3-5]. This work resulted in analytical algorithms that were used to produce the now well-known stability lobe diagram that separates the spindle speed-chip width domain into regions of stable and unstable behavior.

Time domain simulation offers a powerful tool for exploring milling behavior and has been applied to identify instability [6]. For example, Zhao and Balachandran implemented a time domain simulation which incorporated loss of tool-workpiece contact and regeneration to study milling [7]. They identified secondary Hopf bifurcation and suggested that “period-doubling bifurcations are believed to occur” for low radial immersions. They included bifurcation diagrams for limited axial depth of cut ranges at two spindle speeds to demonstrate the two bifurcation types.

The semi-discretization, time finite element analysis, and multi-frequency methods were also developed to produce milling stability charts that demonstrate both instabilities [8]. In [9], it was shown using the semi-discretization method that the period-2 bifurcation exhibits closed, lens-like, curves within the secondary Hopf lobes, except for the highest speed stability lobe. The same group reported further experimental evidence of quasi-periodic (secondary Hopf), period-2, period-3, period-4, and combined quasi-periodic and period-2 chatter, depending on the spindle speed-axial depth values for a two degree of freedom dynamic system [9].

In this paper, period-n bifurcations are experimentally identified for $n = 2, 3, 6, 7$, and 15. Additionally, the sensitivity of the bifurcation behavior

to system dynamics, including both natural frequency and damping, is explored. A comparison of numerical simulation predictions and experiments is presented.

POINCARÉ MAPS

In this study, Poincaré maps were developed using both experiments and simulations. For the experiments, the displacement and velocity of a flexible workpiece were recorded and then sampled once per tooth period. In simulation, the displacement and velocity were predicted, but the same sampling strategy was applied. By plotting the displacement versus velocity, the phase space trajectory can be observed in both cases. The once per tooth period samples are then superimposed and used to interrogate the milling process behavior. For stable cuts, the motion is periodic with the tooth period, so the sampled points repeat and a single grouping of points is observed. When secondary Hopf instability occurs, the motion is quasi-periodic with tool rotation because the chatter frequency is (generally) incommensurate with the tooth passing frequency. In this case, the once per tooth sampled points do not repeat and they form an elliptical distribution. For period-2 instability, on the other hand, the motion repeats only once every other cycle (i.e., it is a sub-harmonic of the forcing frequency). In this case, the once per tooth sampled points alternate between two solutions. For period-n instability, the sampled points appear at n locations.

TIME-DOMAIN SIMULATION

Time-domain simulation entails the numerical solution of the governing equations of motion for milling in small time steps [10]. It is well-suited to incorporating all the intricacies of milling dynamics, including the nonlinearity that occurs if the tooth leaves the cut due to large amplitude vibrations and complicated tool geometries (including runout, or different radii, of the cutter teeth, non-proportional teeth spacing, and variable helix).

Using the time-domain simulation approach, the forces and displacements may be calculated. These results are then once-per-tooth sampled to generate the Poincaré maps.

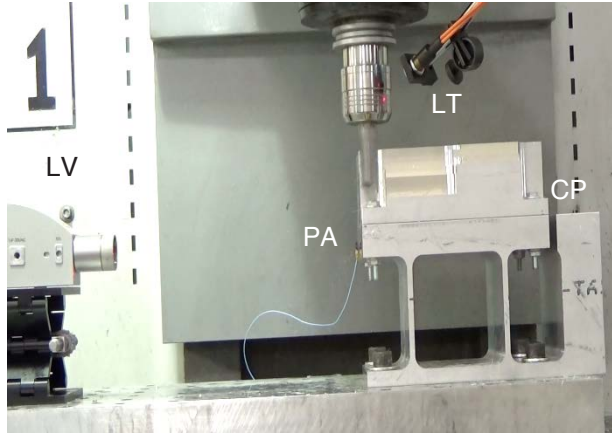


Figure 1. Milling experimental setup with laser vibrometer (LV), piezo-accelerometer (PA), laser tachometer (LT), and capacitance probe (CP).

Table 1. Cutting conditions and flexure dynamics for experiments.

Cutting conditions			
Period-n	Spindle speed (rpm)	Axial depth, b (mm)	Radial depth (mm)
2	3486	2.0	1
3	3800	4.5	5
6	3200	18.0	1
6	3250	15.5	1
7	3200	14.5	1
15	3200	14.0	1
Flexure dynamics			
Period-n	Stiffness (N/m)	Natural frequency (Hz)	Damping ratio (%)
2	9.0×10^5	83.0	2.00
3	5.6×10^6	163.0	1.08
6	5.6×10^6	202.6	0.28
6	5.6×10^6	205.8	0.28
7	5.6×10^6	204.1	0.28
15	5.6×10^6	204.8	0.28

RESULTS

A single degree of freedom (SDOF) flexure was used to define the system dynamics. Modal impact testing verified that the cutting tool dynamic stiffness (1055 Hz natural frequency, 0.045 viscous damping ratio, and $4.2 \times 10^7 \text{ N/m}$ stiffness) was much higher than the

SDOF flexure. The flexure setup also simplified the measurement instrumentation. The flexure motions were measured using both a laser vibrometer and a low mass accelerometer. In order to enable once per tooth sampling of the vibration signals, a laser tachometer was used. A small section of reflective tape was attached to the tool and the corresponding laser tachometer signal used to perform the once per tooth sampling.

The cutting tool was a 20 mm diameter, single flute carbide square endmill. Modal impact testing verified that the cutting tool stiffness was much higher than the SDOF flexure. Each cut of the 6061-T6 aluminum workpiece was performed using a feed per tooth of 0.10 mm/tooth .

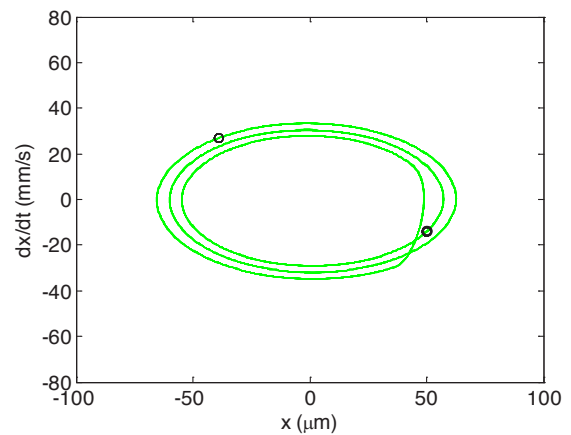


Figure 2(a). Poincaré map for simulated period-2 bifurcation.

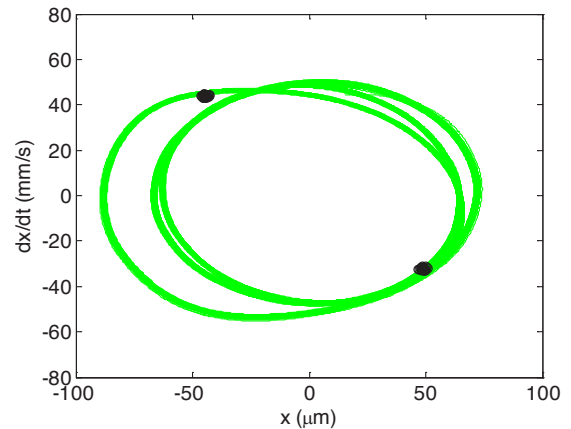


Figure 2(b). Poincaré map for experimental period-2 bifurcation.

Cutting tests were completed using the Fig. 1 setup. The measured flexure dynamics and cutting conditions are listed in Table 1. Results for period-2, 3, 6, 7, and 15 bifurcations are displayed in Figs. 2-7. In each figure, (a) shows the simulated behavior and

(b) shows the experimental result. Good agreement is observed in each case.

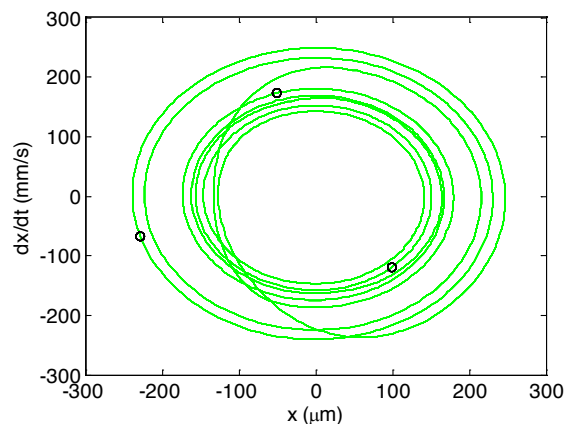


Figure 3(a). Poincaré map for simulated period-3 bifurcation.

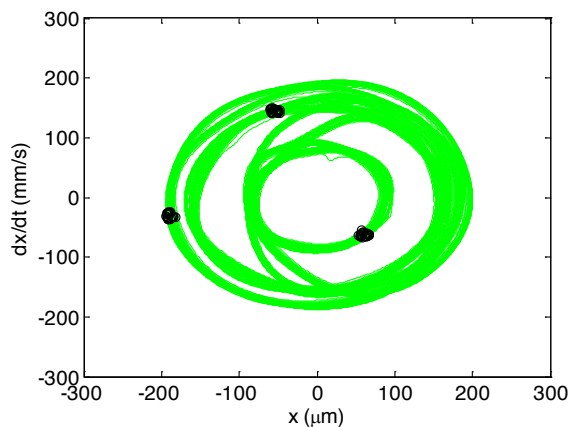


Figure 3(b). Poincaré map for experimental period-3 bifurcation.

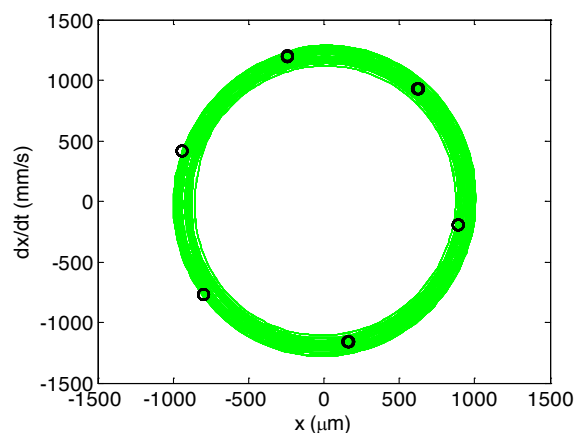


Figure 4(a). Poincaré map for simulated period-6 bifurcation.

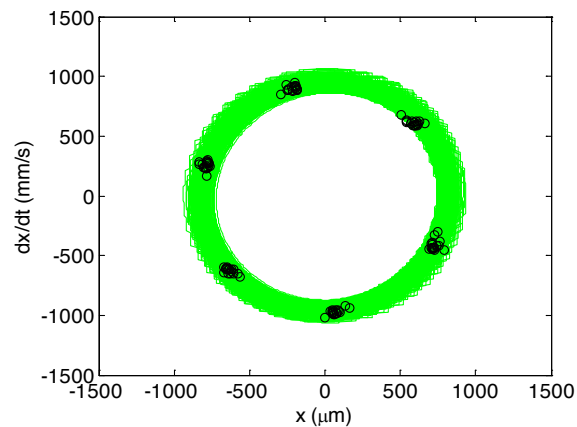


Figure 4(b). Poincaré map for experimental period-6 bifurcation.

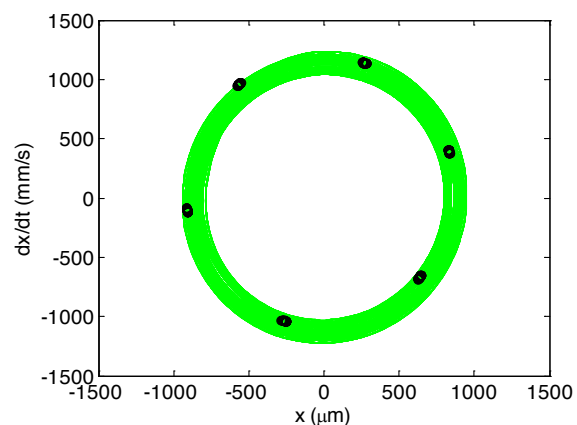


Figure 5(a). Poincaré map for second simulated period-6 bifurcation.

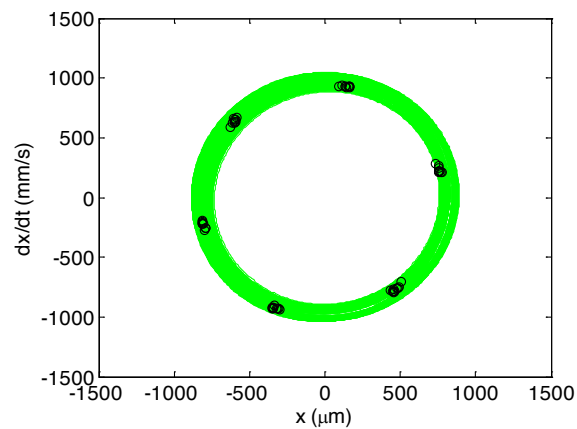


Figure 5(b). Poincaré map for second experimental period-6 bifurcation.

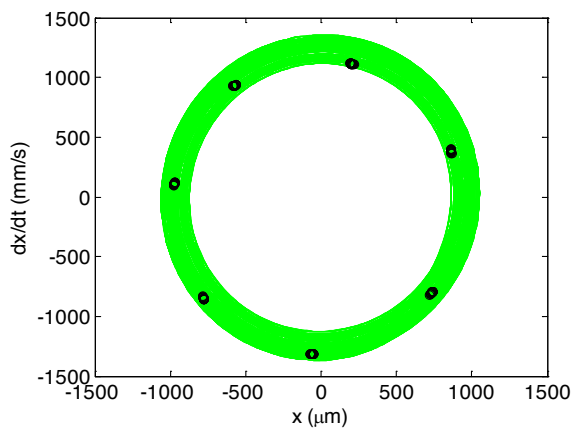


Figure 6(a). Poincaré map for simulated period-7 bifurcation.

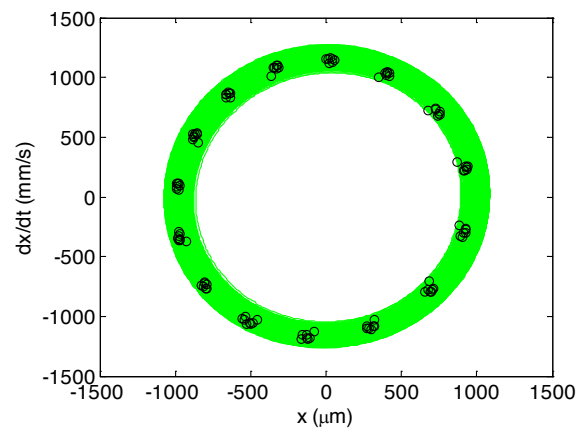


Figure 7(b). Poincaré map for experimental period-15 bifurcation.

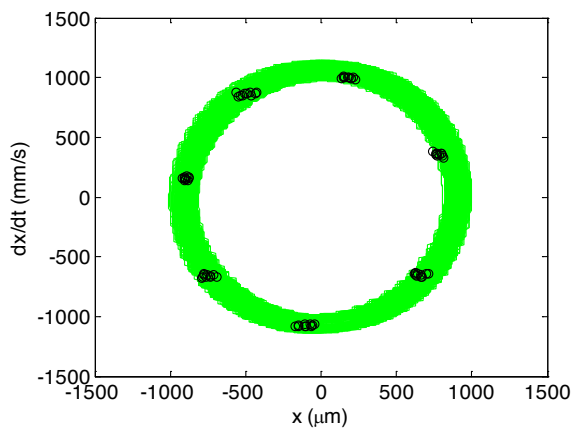


Figure 6(b). Poincaré map for experimental period-7 bifurcation.

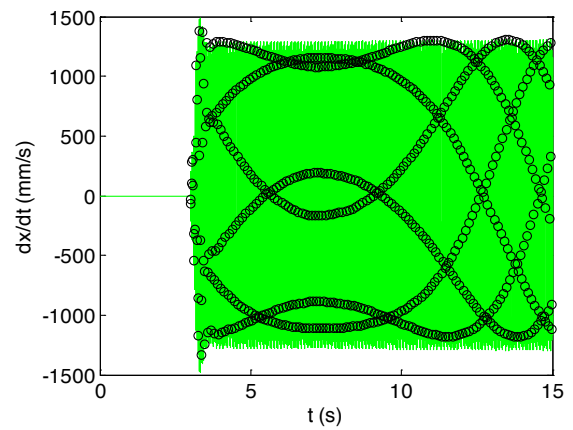


Figure 8(a). Variation in bifurcation behavior with changes in natural frequency for simulated period-6 bifurcation.

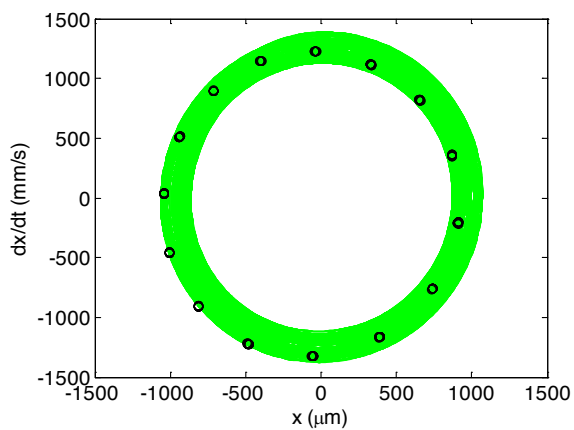


Figure 7(a). Poincaré map for simulated period-15 bifurcation.

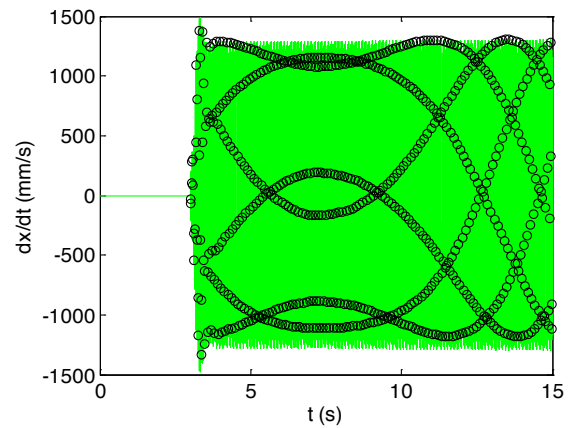


Figure 8(b). Variation in bifurcation behavior with changes in natural frequency for experimental period-6 bifurcation.

Experiments were also completed to demonstrate the sensitivity of the period- n bifurcation behavior to changes in natural frequency. During cutting, material is removed from the workpiece which lowers the workpiece mass and, subsequently, increases the flexure's natural frequency. Since the mass of the chips is much smaller than the workpiece, these changes result in small changes in natural frequency. The changes in system dynamics for the experiments presented in Figs. 8-11 are provided in Table 2. The higher period- n bifurcations exhibited sufficient sensitivity to flexure natural frequency that, within a single cut, both period- n bifurcation and quasi-periodic behavior were observed.

Table 2. Changes in flexure natural frequency due to mass removal.

Flexure dynamics				
Period - n	Start natural freq.(Hz)	End natural freq. (Hz)	Natural freq. change (Hz)	Mass loss (g)
6	202.4	202.7	0.3	4.8
6	205.7	205.9	0.2	4.1
7	204.1	204.3	0.2	3.9
15	204.7	204.9	0.2	3.7
Cutting conditions				
Period - n	Spindle speed (rpm)	Axial depth, b (mm)	Radial depth (mm)	
6	3200	18.0	1	
6	3250	15.5	1	
7	3200	14.5	1	
15	3200	14.0	1	

Figures 8-11 show the flexure's feed (x) direction velocity (dx/dt) versus time. The continuous signal is displayed as a solid line, while the circles are the once-per-tooth sampled points. In each figure, (a) shows the simulated behavior and (b) shows the experimental behavior. Good agreement is observed. The time-domain simulation was altered to account for the changing natural frequency due to mass loss. After each time step, the change in mass was calculated based on the volume of the removed chip and the density of the workpiece material. This change in mass was then used to update the flexure's natural frequency for the next time step.

A summary of the behavior seen in Figs. 8-11 is provided here.

1. Figure 8 exhibits period-6 behavior from 4 to 11 s, followed by quasi-periodic behavior until the end of the cut.
2. Figure 9 shows period-6 behavior from 4 to 13 s and then quasi-periodic behavior is observed until the end of the cut.
3. Figure 10 displays quasi-periodic behavior from the beginning of the cut until 11 s and then period-7 behavior from 11 to 15 s.
4. Figure 11 exhibits quasi-periodic behavior from the beginning of the cut until 8 s, period-15 behavior from 8 to 13 s, and then quasi-periodic behavior until the end of the cut

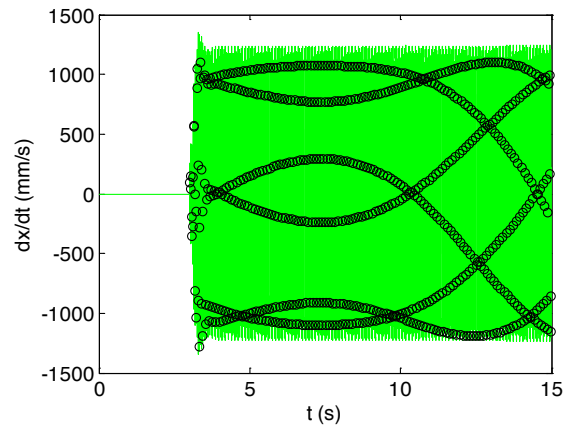


Figure 9(a). Variation in bifurcation behavior with changes in natural frequency for second simulated period-6 bifurcation.

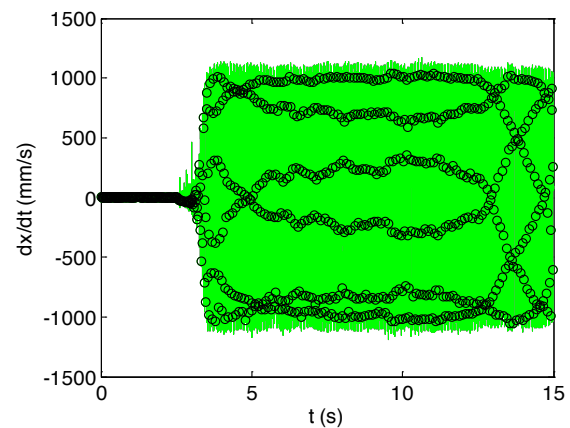


Figure 9(b). Variation in bifurcation behavior with changes in natural frequency for second experimental period-6 bifurcation.

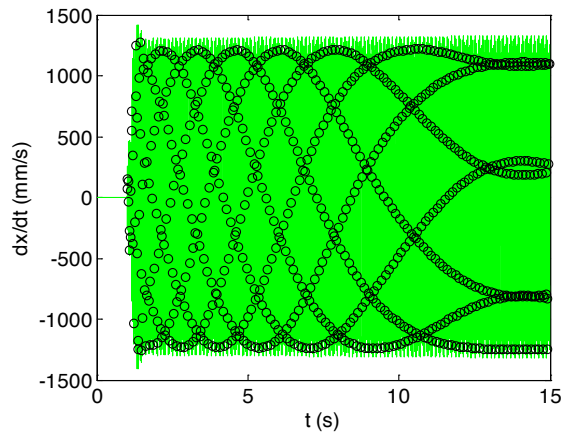


Figure 10(a). Variation in bifurcation behavior with changes in natural frequency for simulated period-7 bifurcation.

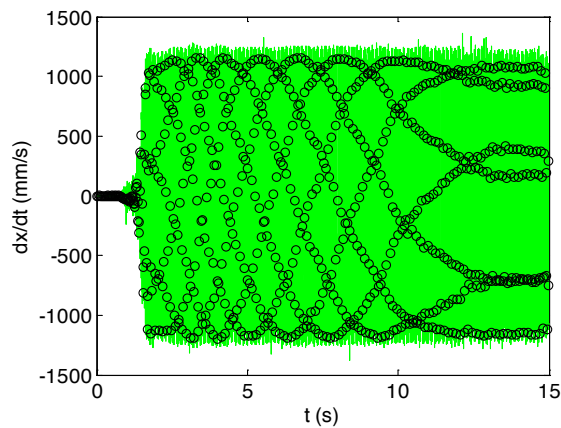


Figure 10(b). Variation in bifurcation behavior with changes in natural frequency for experimental period-7 bifurcation.

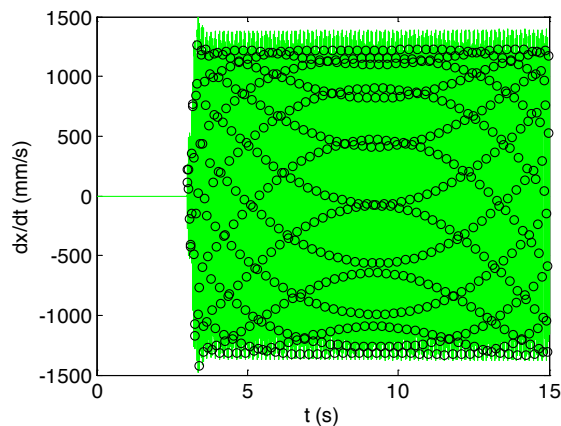


Figure 11(a). Variation in bifurcation behavior with changes in natural frequency for simulated period-15 bifurcation.

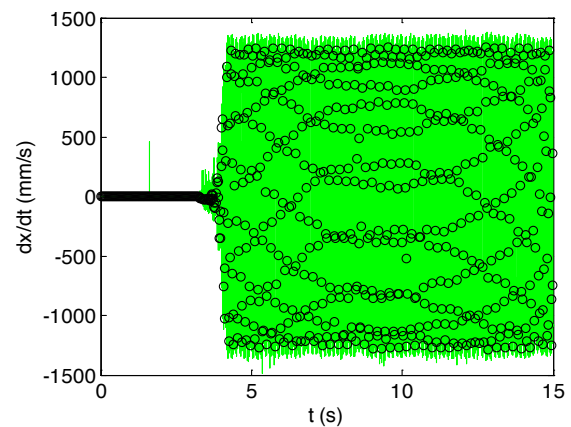


Figure 11(b). Variation in bifurcation behavior with changes in natural frequency for experimental period-15 bifurcation.

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A study on micro machining for super elasto-plastic titanium alloy with small ball end mill tool -Influence of cutting temperature and cutting force on affected layer-

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1. Introduction

Recently, the demand for artificial replacement surgery in the dental and medical fields has increased as a result of the aging society of Japan and the advancement of medical technology. As implant materials, Co–Cr alloys and stainless steel have been commonly used, but these alloys contain elements that may have highly cytotoxic effects. Thus, titanium alloys have been applied more recently because of their excellent mechanical properties and biocompatibility^[1]. In the Japanese Industrial Standards (JIS) T 7412, six titanium alloys, including pure titanium, have been approved as orthopedic implant materials^[1]; however, the Young's moduli of these materials are approximately 100 GPa. In contrast, the cortical bone in which such implants would be buried have Young's moduli of 10–30 GPa, meaning the bone resorption is accompanied by stress shielding due to the difference between these Young's moduli. Under this influence, there may be problems with the implant loosening or an increase in the risk of bone fracture as a result of bone thinning^[2]. Accordingly, a super elasto-plastic titanium alloy^[3] has been proposed as a new implant material. This alloy has special properties combining a high tensile strength of approximately 1000 MPa and a low Young's modulus of approximately 40–60 GPa after cold swaging at reduction of area of over 90% on the final of material manufacturing process. Additionally, all of the elements comprising this alloy have extremely low cytotoxicity. Thus, this alloy has ideal characteristics for an implant material because of its excellent mechanical and chemical biocompatibility. The Material form of

TABLE 1. Chemical composition of workpiece material

Chemical composition (mass%)				
Nb	Zr	Ta	O	Ti
36.3	2.9	2.0	0.31	Bal.

TABLE 2. Mechanical properties of workpiece material

Tensile strength (MPa)	Young's modulus (GPa)	Vickers hardness (HV)	Elongation (%)
980	62	255	12

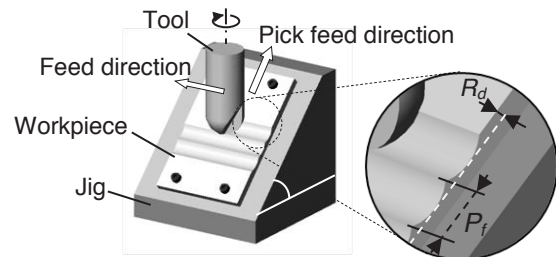


FIGURE 1. Schematic illustration of cutting method

this alloy becomes small because of manufacturing by cold swaging, expecting this alloy to be used for small implants, such as dental implants. Therefore, it is necessary to finish machining such devices with a small cutting tool. However, this alloy is a β -type titanium alloy, which reduces the tool life because of its low thermal conductivity and high welding ability during cutting. In a previous study, using a coated tool was shown to extend tool life, and AlCrSiN coating was found to be the most

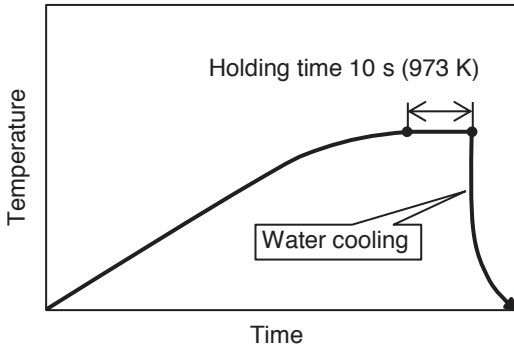


FIGURE 2. Heat treatment chart for workpiece material

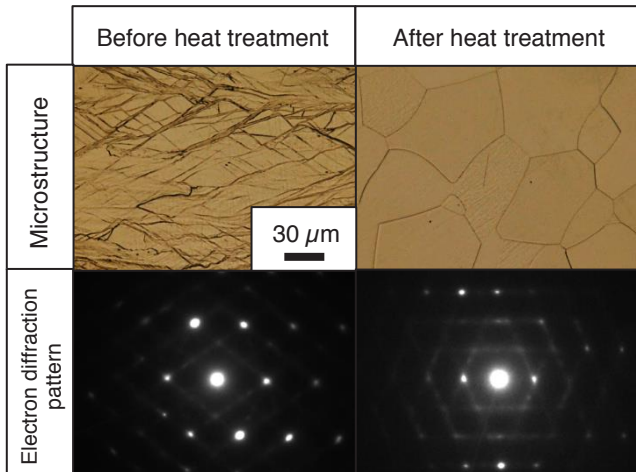


FIGURE 3. Comparison of the microstructures and electron diffraction patterns of the workpiece before and after heat treatment

effective coating for extending tool life^[4]. However, the low Young's modulus of the titanium alloy, which is an important property for implant materials, is lost in high-temperature environments or under high forces^[5]. A high-temperature, hyperbaric environment exists around the cutting position; thus, it is possible for the low Young's modulus of the alloy to be lost during machining. This necessitates removing the affected layer of the alloy workpiece to observe the mechanical properties. In this study, the influence of the cutting temperature and cutting force on the affected layer was investigated by milling using a small ball end mill.

2. Experimental method

A β -type super elasto-plastic titanium alloy was used as the workpiece in the present experiment. The chemical composition and mechanical properties of the workpiece are

given in Tables 1 and 2, respectively. This material can achieve a high tensile strength and a low Young's modulus to undergoing cold swaging with a 90% reduction in surface area. An AlCrSiN-coated ball end mill tool was used as the machine tool. The substrate material was cemented carbide, and the coated film thickness was approximately 1.0 μm . The outer diameter of the tool was 1.0 mm, and the shank diameter was 4.0 mm. The experiment was conducted using a vertical machining center with a high-speed motor spindle, and cutting was performed using the method shown in Figure 1. The cutting speed V was set to 1.0, 1.5, 2.5, and 4.0 m/s; the feed rate S_z was 6 $\mu\text{m}/\text{tooth}$; the radial depth R_d of the cut was 50 μm ; and the pick feed P_f was 30 μm . The cutting environment was wet, and downcutting was performed. Moreover, a water-soluble emulsion with a dilution of 20 times was used and was external supplied at a pressure of 0.5 MPa. The thickness of the affected layer was measured using transmission electron microscopy (TEM). Film samples were produced by cutting the machined workpiece at each considered cutting speed. The workpiece was coated with a protective film of tungsten and carbon and then thinned stepwise at removing the bottom and both side walls of the workpiece using a focused ion beam (FIB) processing apparatus.

3. Results and discussion

3.1 Transformation of material organization by static heating

As mentioned previously, it has been reported that the low Young's modulus of the super elasto-plastic titanium alloy is lost in a high-heat environment or when a large external force is applied. Thus, the change in its microstructure after heat treatment was investigated. Figure 2 shows the heat treatment chart for the workpiece material. Heat treatment was applied at 973 K for 10 s, and water cooling was conducted afterward. During the actual cutting experiment, the cutting environment was wet, so water cooling was applied on the heat treatment to simulate the cutting environment. Figure 3 compares the microstructures and the electron diffraction patterns before and after heat treatment. Before heat treatment, the workpiece had a "marble-like" structure that is particular to the super elasto-plastic titanium alloy^[6]. In contrast, the microstructure after heat treatment contained grainy particles, and the equiaxial crystal grain changed, demonstrating that the microstructure of the material is

transformed by heat treatment. Furthermore, the microstructure after heat treatment was remarkably similar to that before cold swaging^{[3],[6]}, and it has possibility that the mechanical properties changed to the conditions before cold swaging. Comparing the electron diffraction patterns before and after heat treatment shows that the crystal structure also changed as a result of heat treatment; the electron diffraction pattern before heat treatment formed a regular rhomboid pattern, and that after heat treatment showed a regular hexagonal pattern. Thus, the electron diffraction pattern was observed at a cross section of the machined surface to evaluate the affected layer after the cutting process.

3.2 Effect of the cutting speed on the thickness of the affected layer

The thickness of the affected layer was observed at different cutting speeds to determine the effect of the cutting speed on the affected layer. Figure 4 shows a TEM image and an electron diffraction pattern of the cross section of the machined surface at a cutting speed of 2.5 m/s. A different layer is observed under machined surface and thickness is about 0.3 μm on the TEM image. Additionally, the electron diffraction patterns at the locations labeled **a** and **b** in Figure 4 differ from those at **c**–**f**; the patterns at **a** and **b** are hexagonal and are similar to the pattern after heat treatment, whereas those at **c**–**f** are rhomboid patterns and are the same as that before heat treatment, as shown in Figure 3. Thus, the thickness of the affected layer can be determined by evaluating the TEM image and electron diffraction patterns, and the thickness of the affected layer in this experiment was estimated to be 0.3 μm at a cutting speed of 2.5 m/s. Based on this result, evaluated the thickness of affected layer at changing cutting speed. Figure 5 shows the relationship between the cutting speed and the thickness of the affected layer at each cutting condition. The thickness of the affected layer tended to increase with increasing cutting speed.

3.3 Measurement of cutting temperature and cutting force at each cutting speeds

The cutting temperature and cutting force were measured at each considered cutting speed in order to investigate the main factor forming the affected layer. The method of measuring the cutting temperature is shown in

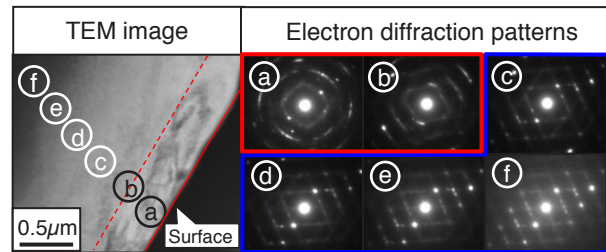


FIGURE 4. TEM image and electron diffraction patterns of a cross section of the machined surface at $V = 2.5$ m/s

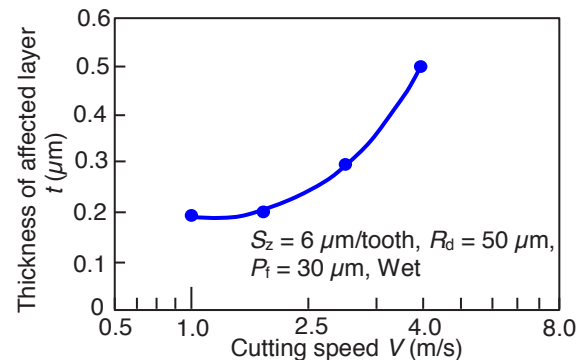


FIGURE 5. Relationship between cutting speed and the thickness of the affected layer at each considered cutting speed

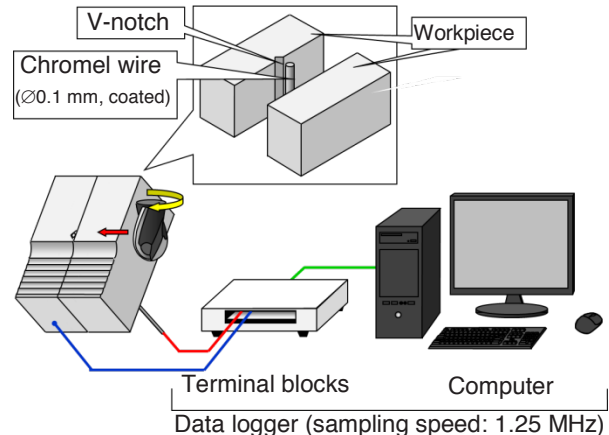


FIGURE 6. Schematic illustration of the method of measuring the cutting temperature

Figure 6. The cutting temperature was measured using a thermocouple between the workpiece and a chromel wire. The chromel wire was coated with a ceramic isolation film and had a diameter of 0.1 mm. Prepare the two workpiece and chromel wire at the V-notch on the workpiece one of pairs and sand with another one workpiece. The thermocouple uses a method in which a closed circuit is formed when the chromel wire is shorted to the

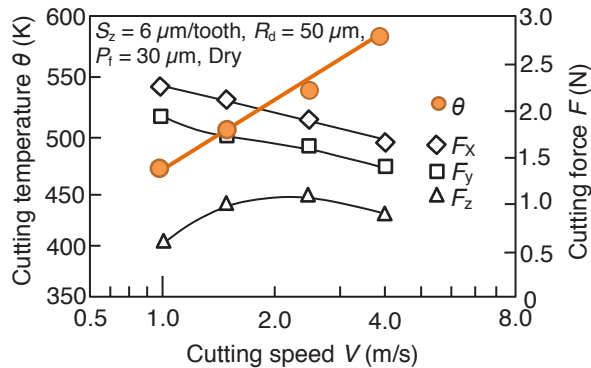


FIGURE 7. Cutting temperature and cutting force plotted against the cutting speed

workpiece by cutting. A thermoelectromotive force was output from the thermocouple by simultaneously cutting the workpiece and chromel wire. This thermoelectromotive force was converted to an average temperature using a calibration curve that had been prepared in advance. The method of measuring the cutting force used a three-component force link (Kistler 9317B).

Figure 7 shows the cutting temperature and cutting force plotted against the cutting speed. The cutting temperature tended to increase with increasing cutting speed, and a maximum temperature of approximately 590 K at a cutting speed of 4.0 m/s was obtained under the given conditions. In contrast, the cutting force tended to have low values and decrease with increasing cutting speed. The maximum cutting force was less than 2.5 N. These results indicate that the affected layer was mainly formed as a result of high cutting temperatures, not high cutting forces.

3.4 Effect of cutting temperature and cutting force on the thickness of the affected layer

Based on the measurement results, the effects of the cutting temperature and cutting force on the thickness of the affected layer were verified. The cutting temperature was approximately 900 K at a cutting speed of 16 m/s. This temperature was significantly higher than those at other cutting speeds. Thus, the thickness of the affected layer at a cutting speed of 16 m/s was inspected to evaluate the effect of the cutting temperature. Figure 8 shows a TEM image and an electron diffraction pattern of a cross section of the machined surface at a cutting speed of 16 m/s. It is a tendency similar in the area of a TEM image. Additionally, the electron diffraction pattern was hexagonal at all

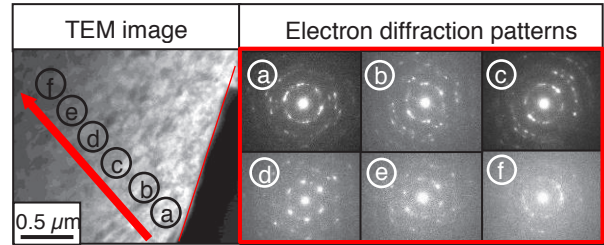


FIGURE 8. TEM image and electron diffraction patterns of a cross section of the machined surface at $V = 16$ m/s

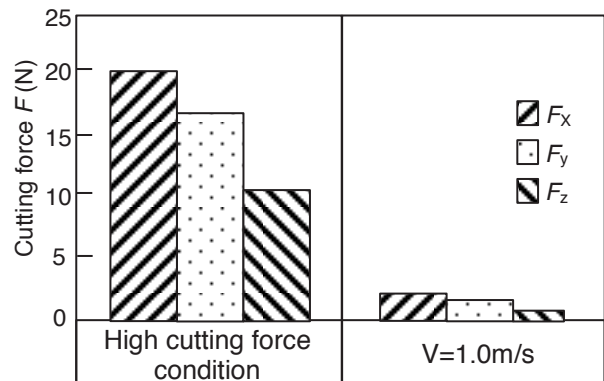


FIGURE 9. Comparison of the cutting forces between under high cutting force condition and at $V = 1.0$ m/s

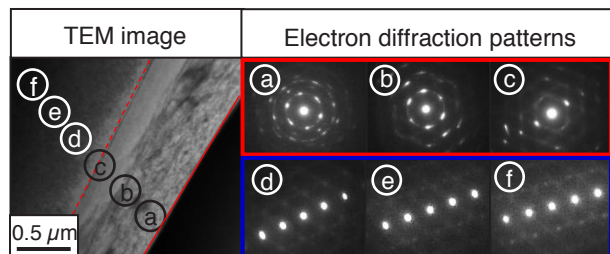


FIGURE 10. TEM image and electron diffraction patterns of a cross section of the machined surface under high cutting force condition

observed points. Thus, the thickness of the affected layer was over $2.5 \mu\text{m}$ at a cutting speed of 16 m/s. The reason for the extreme increase in the thickness of the affected layer is considered to be that the cutting temperature is close to the transformation point of the workpiece, which is approximately 1000 K^[7].

A high-cutting force environment was achieved by using a high depth of cut and a high feed rate; the depth of cut was $100 \mu\text{m}$, the feed rate was $200 \mu\text{m/tooth}$, and the pick feed was $300 \mu\text{m}$. Additionally, the cutting speed was set to 0.5 m/s to reduce the effect of the cutting temperature on the cutting surface. This set of cutting conditions is called "High cutting force

condition". Figure 9 shows a comparison of the cutting forces under high cutting force condition and at a cutting speed of 1.0 m/s. All components of the cutting force at high cutting force condition were 10 times those at cutting speed of 1.0m/s. Figure 10 shows a TEM image and electron diffraction patterns of a cross section of the machined surface under high cutting force condition. Based on the TEM image and the variation in the electron diffraction pattern, the thickness of affected layer was approximately 0.7 μm . The thickness of the affected layer increased more with increasing cutting force than cutting speed of 1.0m/s. However, the thickness of the affected layer at a cutting speed of 16 m/s was extremely thick than that under high cutting force condition. These results demonstrate that the cutting temperature can be used more effectively than the cutting force to control the thickness of the affected layer. Additionally, it is possible that the affected layer was decreasing to machined low cutting temperature.

4. Summary

In this study, the influence of the cutting temperature and cutting force on the Young's modulus of the affected surface layer of a β -type super elasto-plastic titanium alloy workpiece was investigated by milling using a small ball end mill. The obtained results are summarized as follows.

- (1) The thickness of the affected layer tended to increase with increasing cutting speed and was approximately 0.5 μm at a cutting speed of 4.0 m/s.
- (2) As the cutting speed increased, the cutting temperature increased, and the cutting force decreased.
- (3) The thickness of the affected layer exceeded 2.5 μm at a cutting speed of 16 m/s. The factor causing the extreme increase was the cutting temperature approaching the transformation point of this workpiece.
- (4) The formation of the affected layer can be more effectively controlled by varying the cutting temperature more than the cutting force.

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CHARACTERIZING DIMENSIONAL FORM, SURFACE TEXTURE AND FEATURE RESOLUTION IN ADDITIVE MANUFACTURING

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Additive manufacturing (AM) is considered a disruptive technology with the potential to radically alter product design and to introduce paradigm changes across a broad range of industries and applications. The precision engineering and metrology communities are not immune to AM's impact as new techniques and systems are needed to characterize the equipment, processes, materials and parts involved in or produced by AM.

Traditional subtractive manufacturing methods (i.e. machining) generate a desired geometry through removal of stock material using a cutting tool whose location is controlled by the machine's motion platform. To first order, dimensional geometry errors are driven by the accuracy of the machine tool, while tool interactions with the workpiece tend to control the surface finish generated by machining. The location of the tool can be predicted through machine modeling [1,2]. We can express the actual location of the tool as a vector function:

$$\begin{aligned}\mathbf{r}_{act} &= \mathbf{r}_{nom} + \delta \\ \delta &= f(x, y, z)\end{aligned}\quad (1)$$

where $\mathbf{r}$ and δ are vectors, and the subscripts denote actual position or nominal position. Final part geometries are generally considered to be deterministic using a range of metrology techniques developed and refined within the precision engineering community over the previous decades.

Additive manufacturing introduces a paradigm shift wherein the manufacturing system deposits material at a desired location, building onto existing materials and structures. Final part geometry becomes an outcome of the positioning accuracy of material placement, the physics of the deposition and fusing processes at each material location, the deformation from residual material stresses, and the cumulative errors of the build process. As a result, additive deposition process and interactions provide a substantial contribution to geometrical accuracy

which varies with location and orientation within a part volume. While machine characterization and metrology remains crucial for additive process control, it is not adequate or practical for an "a priori" deterministic analysis of geometrical accuracy as in conventional machining processes.

Researchers over the past couple of decades have developed metrology artifacts typically suited to their process, material and/or performance metric of interest [3]. Our prior work [4] paralleled many of these efforts as we proposed a family of artifacts for characterizing process accuracy, form, overhangs, surface texture and minimum feature size. Follow-on work has focused on the use of two of these artifacts; the multi-sided polyhedron and the 3D Siemens star; for quantifying the dimensional accuracy, surface form error, surface texture and minimum feature resolution of printed parts.

The multi-sided polyhedral structure, Figure 1, is relatively straightforward, but is particularly useful in providing information on process anisotropies. It is also easily modified to incorporate multiple aspects relative to size, number of angles or even its internal structure. The explored geometry involves 26 sides incremented successively by 45° with a total part

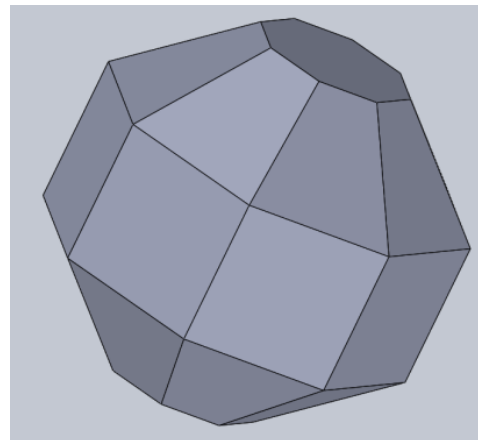


FIGURE 1. 26-sided polyhedron model.

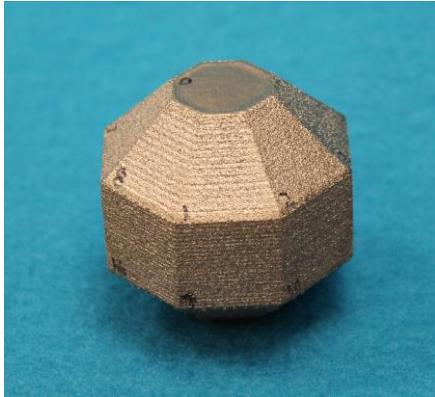


FIGURE 2. Ti6Al4V polyhedron produced by electron beam melting.

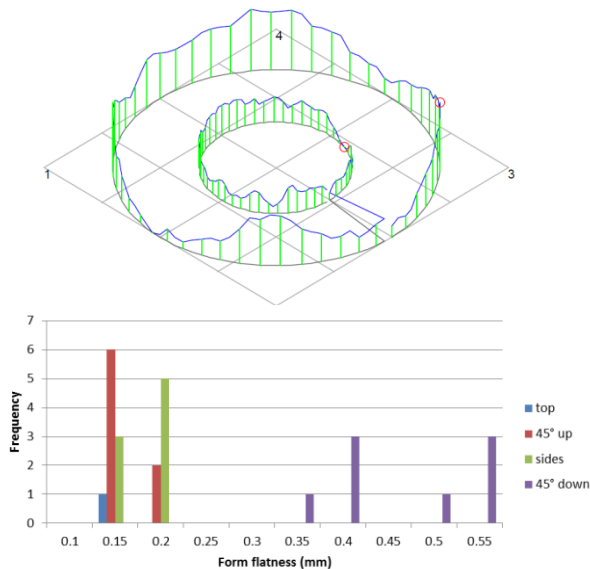


FIGURE 3. Ti6Al4V polyhedron top surface form error (top) and error distribution (bottom).

width of 43 mm. A Ti6Al4V version of this part, Figure 2, was fabricated using electron beam melting (EBM). Each face of the polyhedron was measured using a CMM. Figure 3 demonstrates that surface form accuracy is best for up facing surfaces and degrades with increasing downward facing angles. Dimensional accuracy is quantified by interpolating the intersections between adjacent planar faces and comparing to the nominal design geometry. Form deviations are plotted as vectors, Figure 4, and identify global part shrinkage in all three dimensions.

Additive surfaces, almost irrespective of the process or material, introduce surface texture that is significantly different from traditionally machined surfaces. Figure 5 plots surface metrics for each surface of the Ti6Al6V part. S_a is the surface analog for R_a and is the most

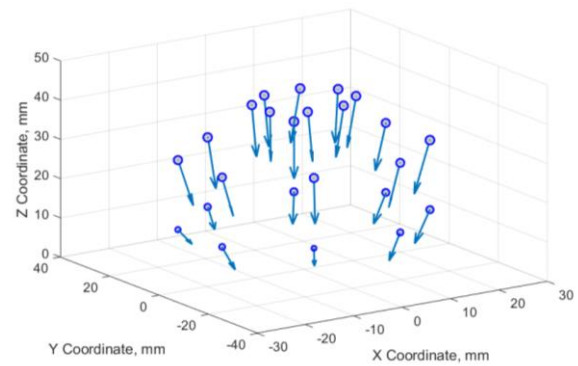


FIGURE 4. Ti6Al4V polyhedron deviation vector plot.

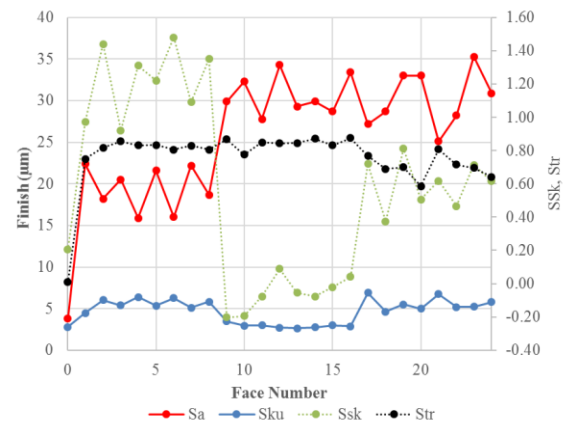


FIGURE 5. Surface finish trends for a Ti6Al4V polyhedron.

common surface metric from profilometry. The surface of the EBM part is less than 5 μm for the top face, but degrades quickly with increasing downward facing angle and approaches 10-25 μm . Other texture metrics are also insightful for additive surfaces. S_{ku} is the surface kurtosis and describes the sharpness of surface peaks. Kurtosis varies slightly across the different polyhedron surfaces, with the top and side surfaces providing the roundest peaks. S_{sk} is surface skewness and differentiates the dominance of peaks or valleys. A positive skewness indicates the presence of more peaks than valleys, a trend observed on the angle surfaces but not necessarily on either the top or sides. S_{tr} is the texture aspect ratio and indicates anisotropy in the texture structure when less than 0.5. Except for the top surface, all other sides of the polyhedron show strong anisotropy that is indicative of the laser scanning performed during part formation. Figure 6 provides representative surface images for the top, 45° up-facing, side and 45° down-facing surfaces highlighting each of these trends.

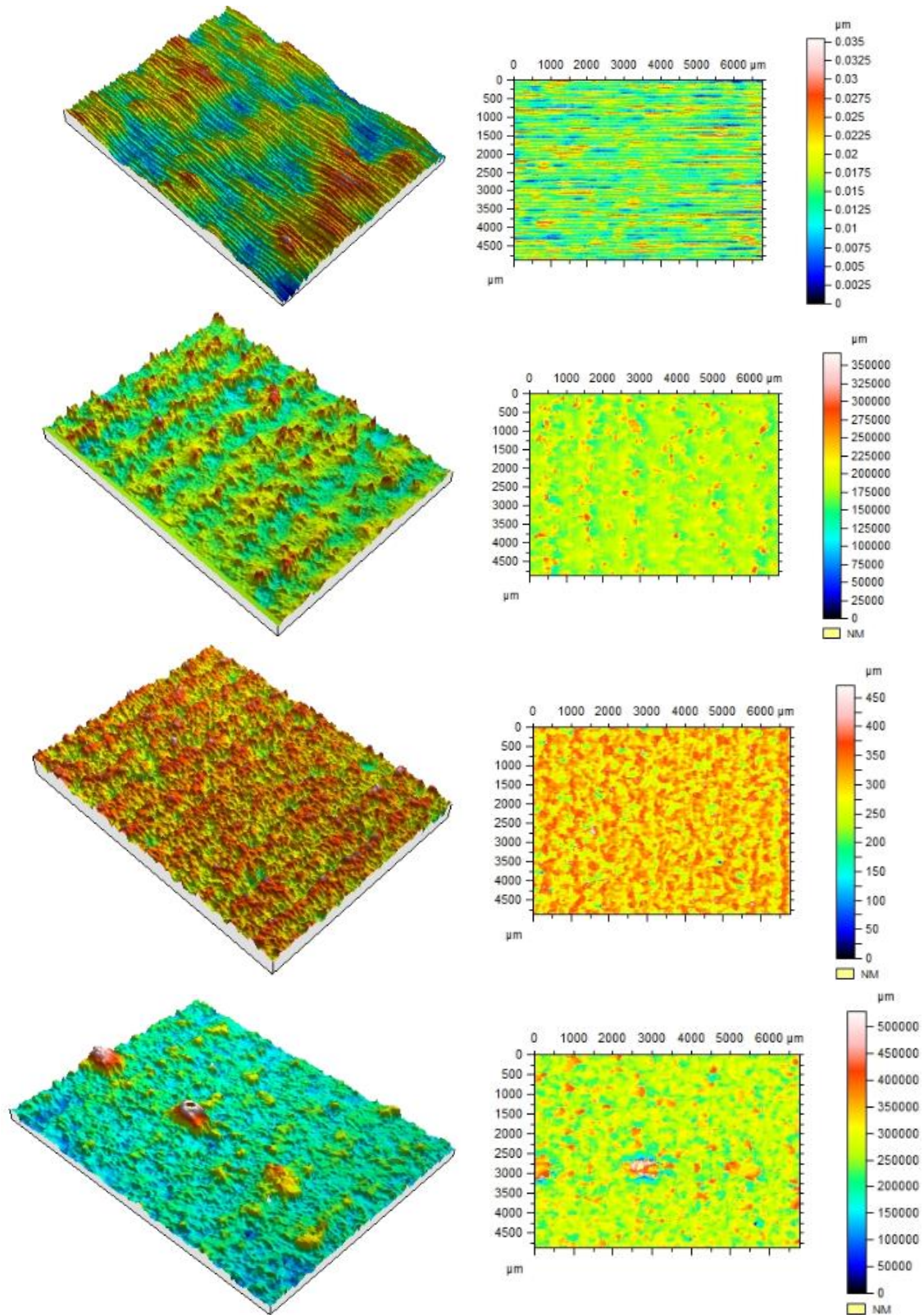


FIGURE 6. Representative Ti6Al4V polyhedron top (first), 45° up-facing (second), side (third) and 45° down-facing (fourth) surfaces.

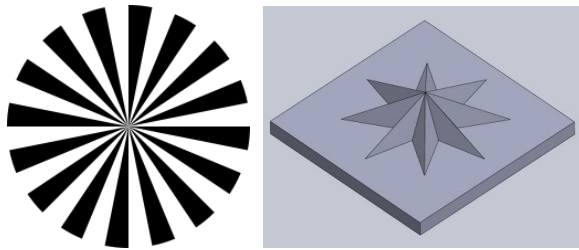


FIGURE 7. 2D (top) and 3D (bottom) Siemens star.

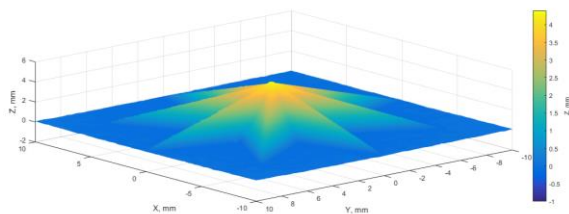


FIGURE 8. 17-4 PH Siemens star produced by laser powder bed fusion (top) with resulting form data (bottom).

The 3D Siemens star is derived from its 2D counterpart [7], Figure 7, which is used extensively to characterize the performance of imaging systems such as optics, displays and 2D printers [5]. Calculating the equivalent of a modulation transfer function (MTF) from the three-dimensional data provides information on the system limits of printable feature sizes. Such information is similar to graphical techniques, i.e. Stedman plots [6], for surface texture instruments and therefore is believed to be useful in the evaluation and comparison of AM equipment, processes, materials and even metrology techniques. Since the geometry is based on simple triangles it can be represented exactly by a STL file, the existing de facto standard for additive equipment and processes. Therefore, errors associated with part representation are eliminated. Figure 8 shows a Siemens star printed in 17-4 PH steel using

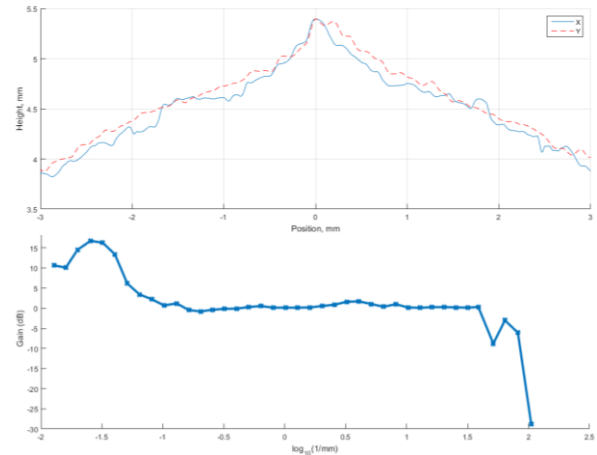


FIGURE 9. 17-4PH Siemens star tip profile (top) and modulation transfer function (bottom)

laser powder bed fusion and its associated form data. Analysis of the star's tip, Figure 9, indicates that combined feature resolution for the machine, process and material are on the order of $100\ \mu\text{m}$ which is consistent with the beam diameter of the process laser. Figure 9 provides the MTF for the part and shows part accuracy degrading at higher and lower length scales.

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Hydrophobic Property of Micro Trench Patterns Machined by Using Diamond Tool and Shaper on Ni-Plated Surface

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INSTRUCTIONS

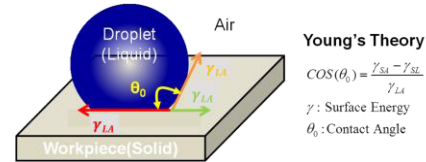
Super-hydrophobic surfaces are characterized by a droplet contact angle greater than 150°. These surfaces have the benefits of being self-cleaning, and preventing contaminants [1]. These surfaces are composed of composite structures that have micro- and nano-sized. The super-hydrophobic surfaces are implemented due to the decrease in surface energy that results from the composite structure.

Hydrophobic surfaces are manufactured by various processes, such as laser machining, the application of anodized aluminum oxide (AAO), chemical coating, micro electro mechanical systems (MEMS), and so on. Ultra-precision machining technologies have been studied to develop a more effective process [2-5].

In this paper, micro-trench patterns were machined by the ultra-precision planing method, which has advantages for molds having large area, and the hydrophobic property of the surface of the micro-trenches was analyzed. The basic surface (mirror surface) of a Ni-plated workpiece was machined by a diamond tool, and the droplet contact angle on this surface was measured. Based on Cassie's theory, the basic size of the micro-trench pattern was designed, and the hydrophobicity was analyzed designed trench patterns of various sizes.

CONTACT ANGLE THEORY

On the plane surface of a material, the contact angle of a droplet can be calculated by Young's theory, as shown in Figure 1 [6]. However, the contact angle can be calculated by Cassie's theory when the droplet is located on the upper surface of fine structures as shown in Figure 2 [7]. According to Cassie's theory, the contact angle on the plane surface and the solid fraction of the structure has an effect on the contact angle on fine structures. The solid fraction can be calculated by dividing the projected-area into the contact area between the droplet and structure.



Young's Theory

$$\cos(\theta_0) = \frac{\gamma_{sa} - \gamma_{sl}}{\gamma_{la}}$$

γ : Surface Energy

θ_0 : Contact Angle

FIGURE 1. Young's theory regarding the contact angle

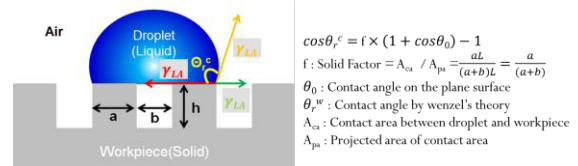


FIGURE 2. Contact angle of Cassie's model

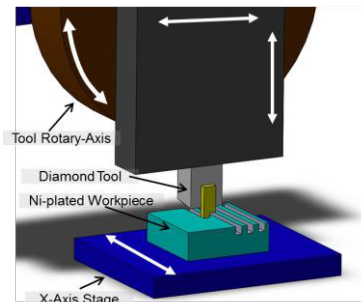


FIGURE 3. Schematic diagram of the 4-axis precision machining system

ANALYSIS OF HYDROPHOBIC PROPERTY OF MIRROR SURFACE OF NI-PLATED WORKPIECE

To design a micro-trench pattern with hydrophobicity, the droplet contact angle on the mirror surface of the material is needed. Thus, an ultra-precision planing method is needed. A schematic diagram of the 4-axis ultra-precision planing system used in this study is shown in Figure 3; each axis has a positioning accuracy of 10 nm. A mirror surface having surface roughness of 11 nm(Ra) was machined on Ni-plated workpiece for droplet contact angle measurement. The specific cutting conditions for the mirror surface are given in Table 1. The

measurement system and conditions are given in Table 2. The measured contact angle result of a droplet placed on the mirror surface was approximately 71°, as shown in Figure 4.

TABLE 1. Machining conditions for mirror surface of Ni-plated workpiece

Items	Conditions
Cutting Tool	10mmR Diamond Tool
Cutting Depth	5 $\mu\text{m}/\text{pass}$
Cutting Pitch	30 μm
Cutting Speed	200mm/sec
Mist Oil	ISOPAR-H

TABLE 2. Measuring system and conditions for droplet contact angle

Items	Conditions
Measuring System	Phoenix300 Touch
Water	DI-Water Density(ρ) : 998kg/m ³
Droplet Volume	5 μL
Drop Speed	5 mm/min

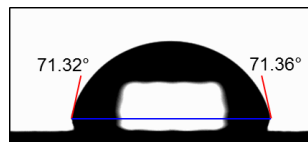


FIGURE 4. Contact angle on the mirror surface of Ni-Plated workpiece

MICRO-TRENCH PATTERN DESIGN BY CASSIE'S THEORY AND EVALUATION OF HYDROPHOBIC PROPERTY

For obtaining a solid fraction in Cassie's theory, the contact angle on the trench surface was assumed to be 150 ° which is standard for super-hydrophobic surfaces. The contact angle on the mirror surface was found to be 71 °. Table 3 shows of the solid fraction and convex-concave width calculated according to Cassie's theory. Also, Table 3 shows the actual applied values; the convex width was decreased from 5.679 μm to 5 μm for convenience of the machining process. A micro-trench pattern which had the shape dimensions of the actual applied values in Table 3 was machined by the previous machining system, and the machining conditions are shown in Table 4. Figure 5 shows a cross-sectional view of the machined micro-trench pattern; a sub-micron dimensional error occurred in comparison to the actual applied values shown in Table 3. Figure 6 shows the droplet contact angle in cross-section direction of this trench pattern; it was approximately 148 °. The phenomenon that the measured contact

angle was lower than the design value was assumed to be caused by fine size error of the machined pattern.

TABLE 3. Values calculated by Cassie's theory and actual applied values of solid fraction, convex-concave width and contact angle

Values Calculated by Cassie's Theory			
Solid fraction	Convex width(μm)	Concave width(μm)	Contact Angle(°)
0.102	5.679	50	150
Actual Applied Values			
Solid fraction	Convex width(μm)	Concave width(μm)	Contact Angle(°)
0.091	5	50	151.57

TABLE 4. Machining conditions for micro-trench pattern designed by Cassie's theory

Items	Conditions
Cutting Tool	Rectangular Diamond Tool (width 50 μm)
Total Cutting Depth	20 μm
Cutting Pitch	55 μm

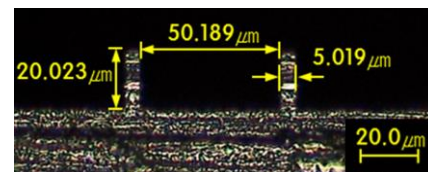


FIGURE 5. Measurement of the machined result of designed micro-trench based on Cassie's theory

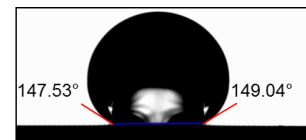


FIGURE 6. Contact angle of droplet on the micro-trench surface of convex width 5 μm and concave width 50 μm

COMPARISON OF THE CONTACT ANGLE WITH VARYING CONVEX AND CONCAVE WIDTH OF THE TRENCH PATTERNS

The effects of varying the size of the designed micro-trench pattern based on Cassie's theory shown in Table 5 were analyzed. The cross-sections of the machined micro-trench patterns are shown in Figure 7, and the machining conditions for these patterns are shown in Table 6. The trench pattern shown in Figure 7 (a) was machined with the shape dimensions of items A in Table 6. Also, Figure 7 (b) was machined with the shape dimensions of items B. The droplet contact angle was measured four times on each upper surface of the micro-trenches, and the

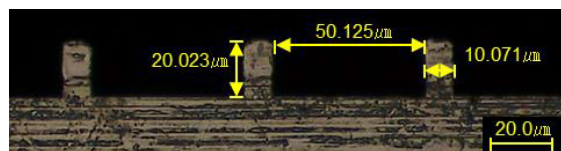
measurement conditions are given in Table 2. The measured contact angle results in the cross-section direction of the trench patterns are shown in Figure 8. Figure 8 (a) shows a contact angle of approximately 140° , while (b) shows a contact angle of approximately 128° . Figure 9 shows comparison graphs of results measured and calculated by Cassie's theory. The measur-

TABLE 5. Shape dimensions of the micro trenches for analysis of hydrophobicity with various concave and convex widths

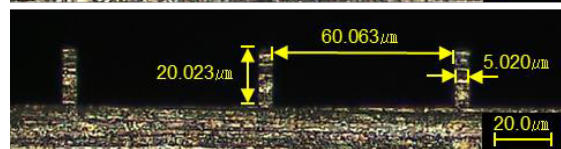
Items	Convex width(μm)	Concave width(μm)	Height (μm)
A	5	50	20
	10		
	15		
	20		
B	5	30	20
		40	
		50	
		60	

TABLE 6. Machining conditions for micro-trench patterns of various sizes

Items	Conditions
Cutting Tools	Rectangular Diamond Tools (width : 30, 40, 50, 60 μm)
Total Cutting Depth	20 μm
Cutting Pitch	Tool Width + Convex Width

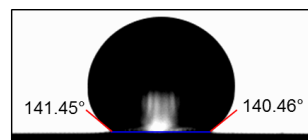


(a) Various convex widths of micro trench

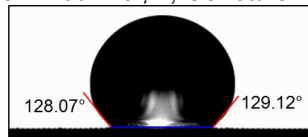


(b) Various concave widths of micro trench
FIGURE 7. Machined micro-trench patterns with various convex and concave widths

ed contact angles showed a tendency to decrease with increasing convex width. This decrease was assumed to be caused by increased contact area between the droplets and the upper surfaces of the micro-trenches. The measured and theoretical contact angle values were similar; the difference between the two values is considered to be due to the shape errors of the trench patterns. The droplet contact angles measured in cross-section direction on the micro-trenches which were machined with the shape dimensions of item B in Table 5 are shown in Figure 10. The contact angles are approximately 144° in Figure 10 (a) and 152° in Figure 10 (b). Figure 11 shows comparison graphs measured and calculated by Cassie's theory. The measured and calculated contact angles showed similar tendency when the trenches had the same convex width and different concave widths. When the concave width was narrow, contact angle decreased. This phenomenon is assumed to be affected by flat surface property when the concave width was narrower.



(a) Convex width 10 μm , Concave width 50 μm



(b) Convex width 20 μm , Concave width 50 μm

FIGURE 8. Droplet contact angle measured in the cross-section direction on micro trenches having the dimensions presented in TABLE 5 A items

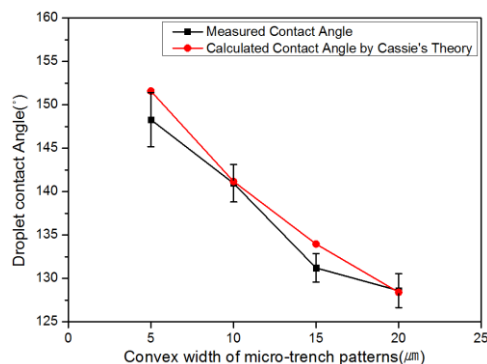


FIGURE 9. Comparison graph of measured contact angle with increasing convex width of the trench patterns and contact angle calculated by Cassie's theory

The results of this experiment showed that the increment of the convex width can increase hydrophilicity of the micro-trench surface. Also, the increment of the concave width can increase the hydrophobicity of the surface. Thus, shape dimension error of micro-trench patterns must be minimized for the realization of designed contact angle on a hydrophobic surface. Also, size optimization of micro trenches is required for the manufacture of a hydrophobic surface through an efficient machining process.

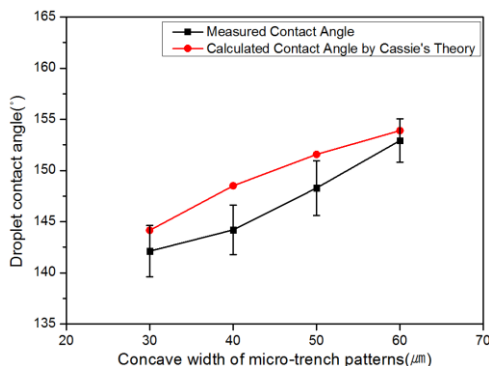
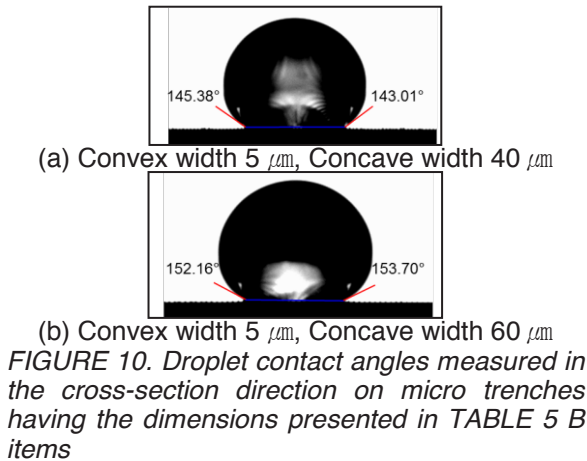


FIGURE 11. Comparison graph of the measured contact angle with increasing concave width of trench patterns and contact angle calculated by Cassie's theory

Conclusion

In this study, micro-trench patterns were machined by a precision planing method using diamond tools. A hydrophobicity assessment was performed by measurement of droplet contact angles on the surfaces of the machined trench patterns. The results are summarized as follows.

- 1) A mirror surface having a surface roughness of 11 nm(Ra) was machined, and its droplet contact angle was approximately 71° .
- 2) Based on Cassie's theory, a trench pattern was designed; convex width of 5 μm , concave

width of 50 μm . The droplet contact angle on this trench surface was approximately 148° ; it was slightly smaller than the intended design due to size-error which occurred during machining.

3) When the concave width was fixed and the convex width was increased in micro-trenches, the droplet contact angle decreased because of the increasing contact area between the droplets and micro-trench patterns.

4) When the convex width was fixed and concave width was decreased in micro-trenches, the contact angle decreased because the effect of the flat surface characteristic increased.

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Study on generation of surface-hardened layer in precision cutting of austenitic stainless steel

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INSTRUCTIONS

Recently, manufacturing industries of industrialized countries are facing a severe business environment due to globalization of markets, shortening of product cycle and low-price competition with companies in the emerging countries. In such critical situations, small and medium-sized manufacture companies suffer economic damages. Especially, in the general purpose parts manufacturing, their manufacturers can be highly inefficient and unprofitable. Therefore, Japanese manufacturing companies began to produce high-value-added parts by improvement of manufacturing processes of difficult-to-cut materials. The machined parts of stainless steel which is one of difficult-to-cut materials are widely used instead of other metals in many industries. In this study, we investigated about production of high-value-added machined surface.

Surface modification of stainless steel has studied from several perspectives, for example, surface strengthening mechanism of work-hardening [1, 2], miniaturization of crystal grain [3], compressive residual stress [4], stress-induced martensite transformation and so on [5, 6]. However, there seems to be few articles explaining relationship between work-hardening and stress-induced martensite transformation in milling process. There are two theories about the mechanism of martensite transformation. Olson et al [7] clarified the theory of nuclear generation of plastic deformation, and Suzuki et al [8] and Onodera et al [9] clarified the theory of stress-induced transformation of local stress concentration. However, there are a lot of issues which don't reveal yet.

This study performed the milling experiment for low-cost and high-value-added machining of

stainless steel AISI304 using one end mill. Newly proposed end milling-burnishing method in this study has been evaluated by the relationship between hardness and generation amount of martensite structure.

Experimental setup and conditions

End milling experiments were carried out with the 3-axis precision cnc vertical machining center as shown in **FIGURE 1**. The cutting tool was 4-flute square TiAlN coated end mill of helix angle 43 degrees. The cutting edge roundness of new tool was about $3\mu\text{m}$. The coolant system was MQL (Minimum Quantity Lubrication). The workpiece was austenitic stainless steel AISI304 which made decarburization and solution heat treatment. The surface hardness before end milling of workpiece by Vickers hardness test (ISO 6507-1:2005) was 163 HV. Experimental apparatus and fixation method of workpiece

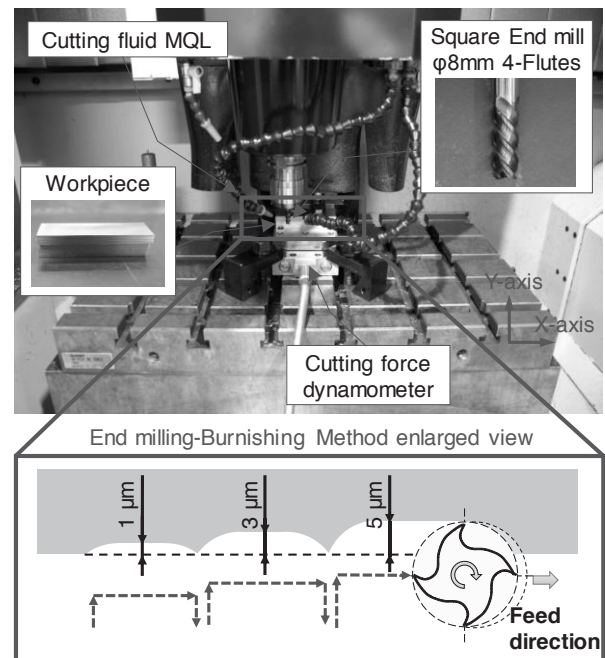


FIGURE 1. Experimental setup

Table 1. Experimental conditions

Machine tool:	Machining center: V33
Cutting tool:	TiAlN coated square end mill Diameter: $D_c = 8$ mm Number of flutes: 4 Helix angle: 43°
Workpiece material:	SUS304
Cutting fluid:	MQL (Vegetable Oil-based Fluid)
Cutting conditions:	Rough milling process: Cutting speed: $V_c = 60$ m/min Feed rate: $f = 50$ $\mu\text{m/tooth}$ Axial depth of cut: $A_d = 8$ mm Radial depth of cut: $R_d = 800$ μm
	EBM process: Cutting speed: $V_c = 20, 60, 100$ m/min Feed rate: $f = 2, 20, 50$ $\mu\text{m/tooth}$ Axial depth of cut: $A_d = 8$ mm Radial depth of cut: $R_d = 1\text{-}5$ μm

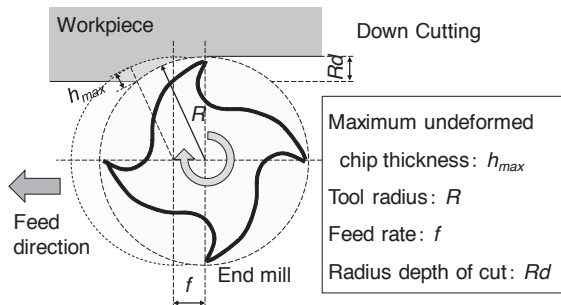


FIGURE 2. Schematic diagram of maximum undeformed chip thickness

material are shown in **FIGURE 1**. All milling experiments were carried out in side cutting by protruding 1 mm from underside of workpiece. In order to facilitate understanding of cutting mechanism, bottom cutting edges of end mill did not use.

End milling-burnishing method proposed in this research can generate the surface modification layer which has compressive residual stress and high hardness by burnishing effect of microscale cutting as shown in **FIGURE 1**. It was necessary to keep the surface condition of workpiece constant before end milling-burnishing method. Then, cutting conditions of rough milling was determined by reference to tool manufacturer recommended milling conditions. Rough milling parameters set to cutting speed $V_c = 60$ m/min, feed rate $f = 50$ $\mu\text{m/tooth}$ and radial depth of cut $R_d = 800$ μm respectively. Main experimental conditions are summarized in **Table 1**. Hardness and crystal structure analysis were evaluated by using micro-Vickers hardness tester and X-ray diffractometer.

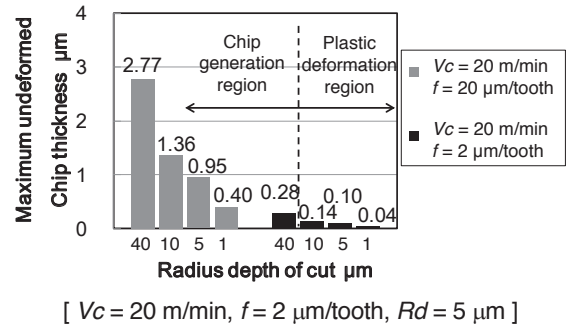


FIGURE 3. Relationship between Maximum undeformed chip thickness and radial depth of cut

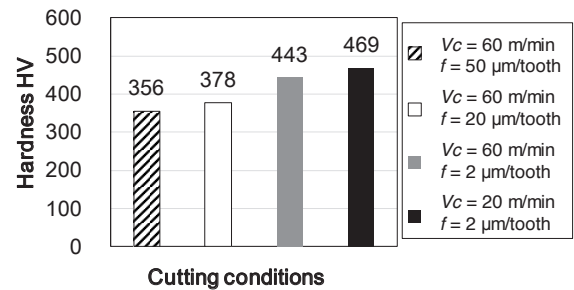


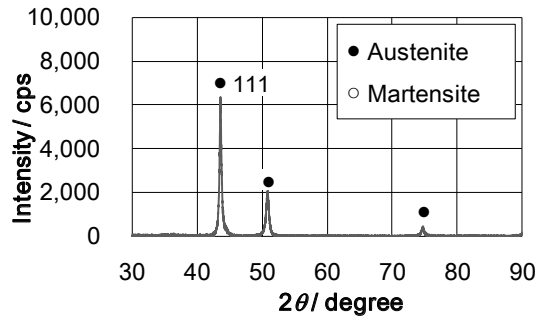
FIGURE 4. Comparison of surface hardness in different cutting conditions

Experimental results and discussion

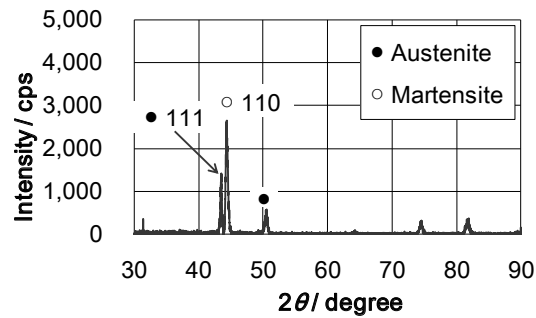
Schematic diagram of end milling and maximum undeformed chip thickness is shown in **FIGURE 2**. Cutter path become trochoid curve because spindle rotation and feed motion are happened at the same time. Maximum undeformed chip thickness is expressed in equation (1).

$$h_{max} = R - \sqrt{R^2 + f^2 - 2f\sqrt{2R * R_d - R_d^2}} \quad \dots(1)$$

Value of Maximum undeformed chip thickness is determined by tool diameter D_c , radius depth of cut R_d and feed rate f . Theoretical values of maximum undeformed chip thickness in several cutting conditions is shown in **FIGURE 3**. In the experiment, chips were not generated under the cutting condition of $V_c = 20$ m/min, $f = 2$ $\mu\text{m/tooth}$, $R_d = 10$ μm . Value of Maximum undeformed chip thickness was 0.14 μm at that time. Therefore, chips were not generated when value of maximum undeformed chip thickness was under 0.14 μm . It is considered that material which should be removed from workpiece is pushed by cutting edge and plastic deformation is occurred. Experimental value of maximum undeformed chip thickness is different from theoretical value because of



[$V_c = 60$ m/min, $f = 50$ $\mu\text{m/tooth}$, $R_d = 5$ μm ,
Down cutting]



[$V_c = 20$ m/min, $f = 2$ $\mu\text{m/tooth}$, $R_d = 5$ μm ,
Down cutting]

FIGURE 5. X-ray patterns of machined surface

difference of cutting edge roundness, machine tool accuracy and so on. Difference of maximum underformed chip thickness has an effect on surface hardness. Comparison of surface hardness in end milling-burnishing process is shown in **FIGURE 4**. Oblique line and white bar show the cutting condition that chips have formed, gray and black bar show the cutting condition that chips have not formed. It is found that surface hardness machined without generating chips is higher than the other machined surface. The reason is hardened layer remain behind on machined surface not to remove as chips. In the case of fixed cutting speed $V_c = 60$ m/min, surface hardness was 356HV under the condition of $f = 50$ $\mu\text{m/tooth}$. Also, surface hardness was 443HV under the condition of $f = 2$ $\mu\text{m/tooth}$. Surface hardness rises as feed rate becomes smaller. In the case of fixed feed rate $f = 2$ $\mu\text{m/tooth}$, surface hardness was 443HV under the condition of $V_c = 60$ m/min. Also, surface hardness was 469HV under the condition of $V_c = 20$ m/min. Surface hardness rises as cutting speed becomes slower.

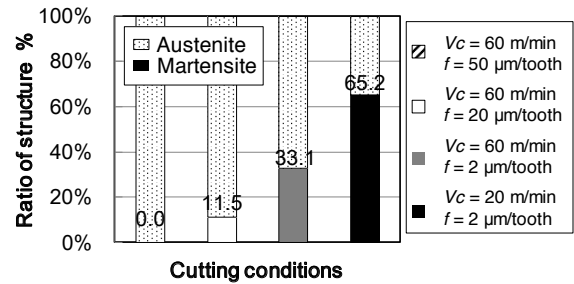


FIGURE 6. Ratio of structures within machined surface

In order to investigate difference of crystal structure within machined surface, machined surface was evaluated by X-ray diffractometer (XRD). Analysis results of machined surface with maximum and minimum hardness are shown in **FIGURE 5**. X-ray diffraction analysis was conducted by $2\theta/\theta$ scan method with CuK- α line. On machined surface with low hardness, austenite peak $\gamma(111)$ only existed around 43.5 degree. It is considered that surface hardness increased with work-hardening caused by plastic deformation. Even if workpiece is machined by end milling-burnishing process in microscale cutting depth, there is the cutting condition which martensite structure do not generate. On the other hand, in the machined surface with high hardness, austenite peak $\gamma(111)$ existed around 43.5 degree and martensite peak $\alpha(110)$ existed around 44.4 degree. Obtained hardness was 469 HV at the highest value. Surface hardness is 2.8 times higher than non-machined surface and 1.7 times higher than machined surface after rough process. Ratio of structures within machined surface is shown in **FIGURE 6**. Machined surface was evaluated by integrated intensity of $\gamma\text{-Fe}$ and $\alpha\text{-Fe}$. Martensite peak ratio is defined in $R = \alpha(110) / [\alpha(110) + \gamma(111)]$. Martensite peak ratio means ratio of martensite structure within machined surface. In a comparison of two different cutting speed, ratio of martensite structure was 33 % under the condition of $V_c = 60$ m/min, $f = 2$ $\mu\text{m/tooth}$, and ratio of martensite structure was 65 % under the condition of $V_c = 20$ m/min, $f = 2$ $\mu\text{m/tooth}$. As feed rate is small, ratio of martensite structure increases. As cutting speed is slow, ratio of martensite structure increases. However, Change of cutting speed is more effectively in consideration of actual machining time.

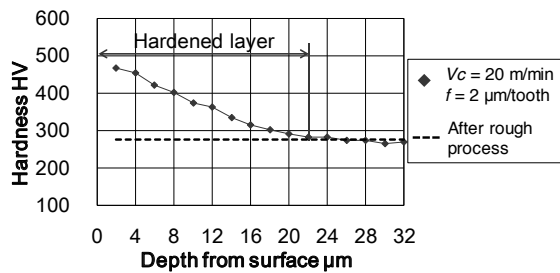


FIGURE 7. Transition of hardness in the depth direction

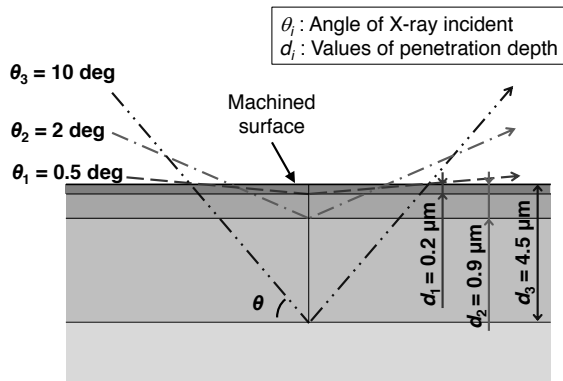
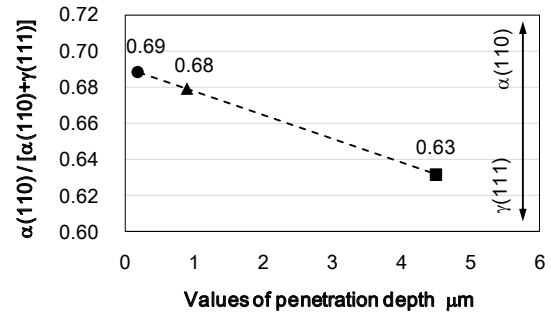


FIGURE 8. Evaluation of machined surface by X-ray diffraction

Measurement results of depth direction hardness test for machined surface, which ratio of martensite structure is the highest, is shown in **FIGURE 7**. Dotted line shows surface hardness 276HV after rough milling. Surface modification layer thickness is defined as the range beyond surface hardness after rough milling. According to obtained results, it is found that thickness of surface modification layer was 22 μm . Schematic diagram of measurement depth by X-ray is shown in **FIGURE 8**. Angles of X-ray incident were installed 0.5, 2, 10 degree to evaluate outer layer of machined surface. According to B.D.Cullity equation [10], the values of penetration depth are 0.2, 0.9 4.5 μm in this incidence angle. Relationship between martensite peak ratio and values of penetration depth is shown in **FIGURE 9**. Values of penetration depth increase from 0.2 to 4.5 μm with decrease of martensite peak ratio from 0.69 to 0.63. It is found that martensite structure increase so as approaching the outer layer of machined surface.



[$V_c = 20 \text{ m/min}$, $f = 2 \text{ } \mu\text{m/tooth}$, $R_d = 5 \text{ } \mu\text{m}$]

FIGURE 9. Relationship between martensite peak ratio and values of penetration depth

CONCLUSIONS

- (1) Change of cutting speed is more effective for generation of martensite structure than that of feed rate.
- (2) Martensite structure existed in abundance at the most outer layer of machined surface. The reason that cutting edges of end mill gave much power for outer layer of machined surface and the distortion of crystal lattice is occurred.

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MODELING AND SIMULATION OF TURNING ON HIGH-GRADIENT ASPHERIC MANDREL WITH AN R- θ MACHINE TOOL

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INSTRUCTIONS

Due to the excellent characteristics of its own profile, the axisymmetric aspheric surface with high steepness has been widely applied in all types of optical systems. In a conformal optical system, it can not only effectively reduce the aircraft's radar cross-section and enhance the performance of the optoelectronic system, but also decrease the air resistance of missiles and other weapon systems, thereby improving their flight speed & range and lowering the air friction heat. In some special space optical and reflective optical systems, it can eliminate central obstruction and correct aberration. In the reflex system of Hard X-ray and Extreme Ultraviolet (EUV), the shape of the reflecting surface is generally designed into a shape of Wolter- I , which is conducive to the collection of light. However, it is quite difficult to manufacture high-precision and high-gradient aspheres. Ultra-precision machine tools with three or more axes are currently used to machine aspheres, but the prices of these machine tools are unacceptable [1,2,3]. For this reason, a Z- θ two-axis synchronous machining method based on an existing ultra-precision machine tools with an R- θ layout was proposed for high-precision and high-gradient axially symmetric aspheres.

The existing lathe of R- θ layout based on swing turning mechanism is shown in Figure 1. Existing ultra-precision machine tools with an R- θ layout use swing feed assisted by cutter radial compensation to machine aspheres and work on workpieces by removing surplus asphericity in the area where the surface profile is most similar to a portion of a sphere. This machining method applies to planes, spheres and aspheres with an asphericity δ smaller than 2.5 mm and is capable of machining workpieces with a maximum aperture of 600 mm; the resulting surface accuracy parameters are as follows: PV= 0.329 λ and RMS=0.029 λ (λ =632.8 nm). However, if an

asphere has a high gradient, its asphericity δ calculated using the aforementioned method will exceed 2.5 mm, making ultra-precision machining of this asphere impossible [4,5].

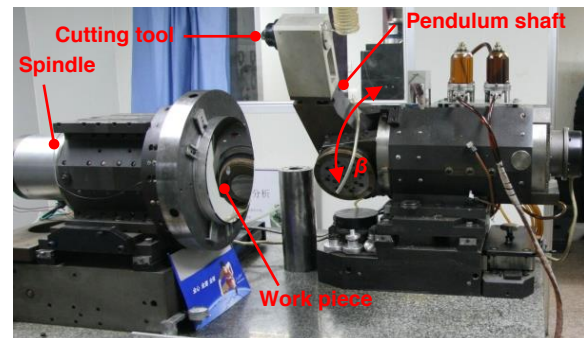


FIGURE 1. R- θ machine layout

PROCESSING MECHANISM AND METHODS

To enable the lathe to process an aspheric surface with high steepness and asphericity, the linkage principle combining the spindle of swing arm rotation (pendulum shaft) and the Z-axis is proposed, as shown in Figure 2. That is, a Z-linear guide of workpiece is added, so that the workpiece can be clamped on the workpiece spindle and perform the controllable reciprocation along the Z-guide; before processing, the axis of rotation and the workpiece spindle are first adjusted to be parallel and equal in height, and then the cutter is fixed at the front end of the swing arm; during processing, the cutter tip performs circular motion along with the swing arm by a swing angle of β , while the Z-guide accordingly performs linear feed motion along with the axis of rotation by a swing angle of β , to accomplish the processing of the aspheric surface with high steepness. Next, the cases where the two spindles (the workpiece spindle and the pendulum shaft) are parallel and intersecting are respectively analyzed.

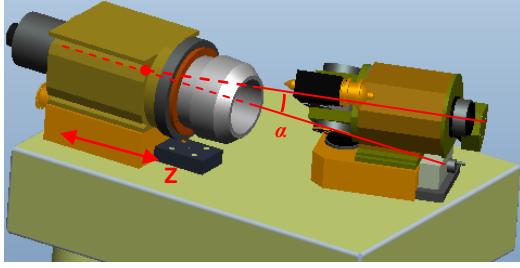


FIGURE 2. The schematic diagram

Parallel Spindles

Before processing, the workpiece spindle and the pendulum shaft are adjusted to be parallel, making the horizontal angle between them into $\alpha=0^\circ$. Based on the meridional profile curve equation of aspheric surface, the coordinate system O_1 is established. Taking the center of pendulum shaft as the origin, the coordinate system O_2 is established as shown in Figure 3. The spacing of the two spindles is P_X , the distance between the cutter tip and the XY plane in coordinate system O_2 is P_Y , and the translation distance of the two coordinate systems along the Z-axis is P_Z . Given that L represents the length of the workpiece, $P_X=430$ mm, $P_Y=(340-L)$ mm, the transformation matrix from coordinate system O_1 to O_2 is expressed as follows:

$$[x_2, y_2, z_2, 1] = [x_1, y_1, z_1, 1] \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ -P_X & 0 & -P_Z & 1 \end{bmatrix} \quad (1)$$

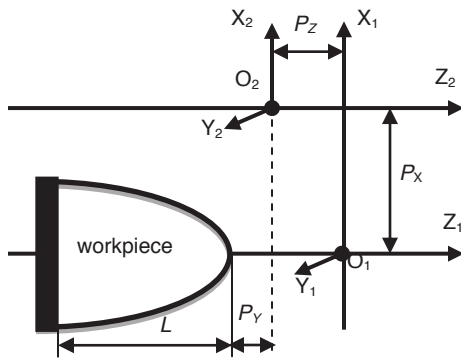


FIGURE 3. Coordinate relation ($\alpha=0^\circ$)

Given that the aspheric surface equation in the coordinate system O_1 is expressed as follows:

$$x^2 = a_1 z + a_2 z^2 + a_3 z^3 + \dots \quad (2)$$

First, according to the dimensions of the processed workpiece, the range of the swing arm's swing angle β is calculated. As the pendulum shaft uniformly swings at an angular velocity of ω , the interpolation points can be roughly selected through equiangular segment of the swing angle β to obtain β_i ($i=1, \dots, n$); when the pendulum shaft uniformly swings at an angular velocity of ω by an angle of β_i , the coordinate values (x_{2i}, y_{2i}, z_{2i}) of the cutter tip in coordinate system O_2 can be calculated as follows:

$$\begin{cases} x_{2i} = P_X \cos \beta_i \\ y_{2i} = P_X \sin \beta_i \\ z_{2i} = -P_Y \end{cases} \quad (3)$$

Obviously, at different values of the swing angle β_i , the corresponding coordinate values (x_{1i}, y_{1i}, z_{1i}) of the cutter tip on the workpiece in the coordinate system O_1 can be calculated by Eq.(3), and thus the motion displacement increment Δz_{1i} , feed velocity $v_f = \Delta z_{1i} / \Delta t$, and acceleration $a_f = \Delta v_f / \Delta t$, ($\Delta t = \Delta \beta / \omega$) of the workpiece along the Z-axis can be calculated.

Intersecting Spindles

When the two spindles are parallel, the range of motion of the workpiece along Z-axis is the length of the workpiece. If an angle $\alpha > 0^\circ$ is maintained between the two spindles, then the workpiece and the cutter move relatively in the z direction. Moreover, the range of motion of the workpiece in the z direction is less than the length of the workpiece, which is conducive to expanding the processing range in the z direction. When the two spindles are parallel, the coordinate systems O_1 and O_2 are established, let coordinate system O_2 rotate counterclockwise around the Y_2 -axis by an angle of α to obtain the coordinate system O_3 , as shown in Figure 4. The spacing between the two spindles is P_X , the distance between the cutter tip and the XY plane in the coordinate system O_2 is P_Y , the translation distance between the coordinate systems O_1 and O_2 in the z direction is P_Z , and R_b is the radius of the base sphere of the cutter tip when the swing arm is swinging. Given that $P_X=430$ mm, $P_Y=[(P_X/\tan \alpha) - R_b]$ mm, when two spindles are parallel, the transformation matrix equation of the coordinate systems O_1 and O_2 is shown in Eq.(1), and the transformation matrix equation of the coordinate systems O_2 and O_3 is shown as Eq.(4).

$$[x_3, y_3, z_3, 1] = [x_2, y_2, z_2, 1] \begin{bmatrix} \cos \alpha & 0 & -\sin \alpha & 0 \\ 0 & 1 & 0 & 0 \\ \sin \alpha & 0 & \cos \alpha & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (4)$$

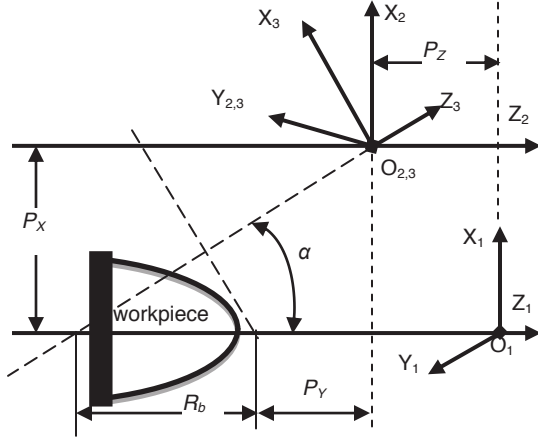


FIGURE 4. Coordinate relation ($\alpha > 0^\circ$)

Similar with the case of parallel spindles, the swing angle β is equally divided. When the axis of rotation uniformly swings at an angular velocity of ω by an angle of β_i , the coordinate values (x_{3i}, y_{3i}, z_{3i}) of the cutter tip in the coordinate system O_3 can be calculated as follows:

$$\begin{cases} x_{3i} = R_b \sin \alpha \cos \beta_i; \\ y_{3i} = R_b \sin \alpha \sin \beta_i; \\ z_{3i} = -[\sqrt{(R_b + P_y)^2 + P_x^2} - R_b \cos \alpha] \end{cases} \quad (5)$$

$$R_b = \frac{P_x}{\tan \alpha} - P_y, \quad 0^\circ < \alpha < \Phi_1$$

$$\Phi_1 = \arctan\left(\frac{P_x}{P_y}\right)$$

Where Φ_1 (deg) represents the critical value of the angle α when processing a convex spherical surface; in this case, $R_b=0$, there is no relative motion between the cutter and the workpiece, and no processing is conducted. Obviously, at different values of the swing angle β_i , the corresponding coordinate values (x_{1i}, y_{1i}, z_{1i}) of the cutter tip on the workpiece in the coordinate system O_1 can be calculated by Eq. (2,4,5).

SIMULATION AND ANALYSIS

Computational simulation is performed on the hyperboloid aspheric surface with high steep-

ness of the optical mandrel, with its meridional profile curve equation as follows:

$$\frac{x^2}{a_2^2} - \frac{y^2}{b_2^2} = 1 \quad (6)$$

Where, $a_2=236.3406$ mm, $b_2=140.5991$ mm. The shape of the mandrel is shown in Figure 5. Where, $\Phi_A=346.4102$ mm, $\Phi_R=444.6342$ mm.

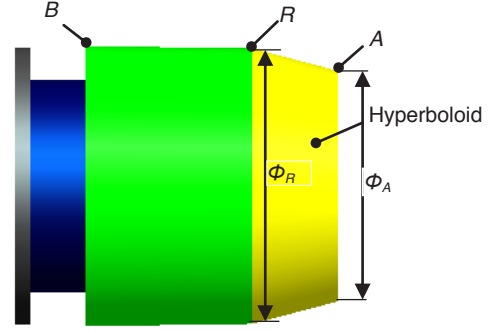


FIGURE 5. The shape of the mandrel

By using the existing principle of the swing feed combining with the cutter radial compensation, the maxim asphericity is calculated as $\delta=3.10$ mm, exceeding the limit of $\delta < 2.5$ mm, and the aspheric surface with high steepness cannot be processed. Therefore, the linkage feed combining the axis of rotation and the Z-axis is adopted to perform the computational analysis before processing. The simulation calculation software is shown in Figure 6, with the calculated results saved in a file of ".txt".

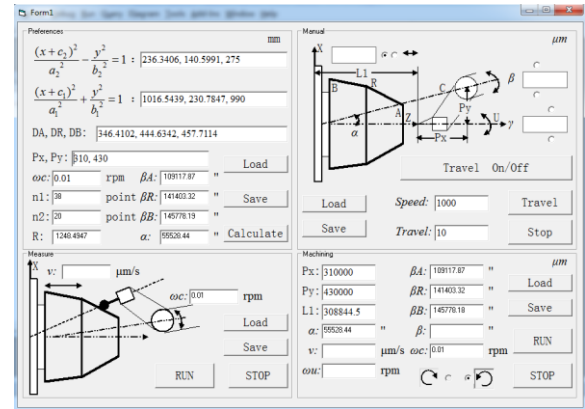


FIGURE 6. Simulation calculation software

Simulation of Parallel Spindles

In this case, $\alpha=0^\circ$, first, it is verified that whether the swing angle of pendulum shaft corresponding to point A and R on the workpiece falls

within the range of 0° - 70° , and the calculated result is $\beta_A=23.2378^\circ$, $\beta_R=29.9631^\circ$, meeting the test requirements. Due to the specific mechanism of processing the aspheric surface with high steepness by swing turning, it is difficult to derive the interpolation error by theoretical formula. Therefore, the meridional profile curve of the processed hyperboloid is divided into n interpolation segments to explore the variation pattern of the interpolation error. The simulated results divided into 1, 2, and 100 interpolation segments, respectively, are shown in Figure 7.

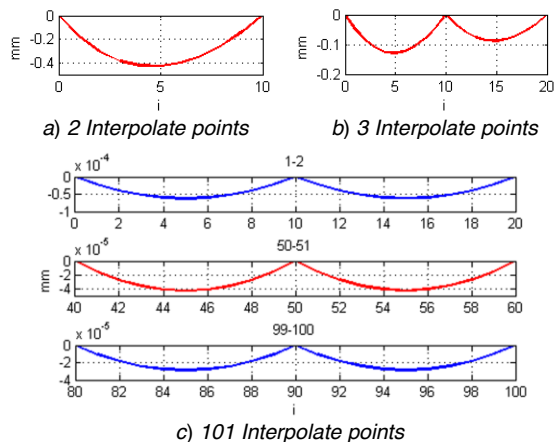


FIGURE 7. The simulated results ($\alpha = 0^\circ$)

According to the simulated results, in theory, the radial error is 0 at the start and end points of each interpolation segment, indicating that this method can ensure that the interpolation points are processed on the curve, but there is radial error in the intermediate points between the start and end points of each interpolation segment. As suggested by the calculated data, the maximum errors of all the interpolation segments are found in their middle parts, and all the radial errors are negative deviation. Comparison of the maximum errors of all interpolation segments shows that the maximum error is found in the first interpolation segment. This is caused by the high steepness of processed workpiece, as well as the greatly varying curvatures at each point on the meridional profile. Because a higher steepness results in more asphericity to be removed, while a smaller radius of curvature leads to a greater chord error. Obviously, the more the interpolation segments, the smaller the surface-shape error. However, limited by the motion resolution of the lathe, the number of the interpolation points cannot be selected infinitely. For this reason, the number of interpolation segments can be deter-

mined according to the processing requirements. For example, if the final surface-shape error is required to be $PV < 1 \mu m$, then the requirement can be met as the number of rough interpolation points is calculated as 26 by the optimization program. In this case, the maximum value of the radial error is found in the first interpolation segment, with a value of $-0.9865 \mu m$.

Simulation of Intersecting Spindles

As mentioned earlier, when there is an angle α existing between the two spindles, both the length of the workpiece and the swing angle of pendulum shaft can be increased, which is helpful to meet the requirements of the swing resolution better. However, since the angle α is limited by the swing range 0° - 70° of the pendulum shaft, an appropriate angle α should be selected to meet the requirements of both swing resolution and range. After several simulation experiments, it is determined that $\alpha=31.2544^\circ$, the radius of corresponding base sphere $R_b=398.4947$ mm, and swing angle $\beta_A=50.8945^\circ$, $\beta_R=68.3976^\circ$ are all within the swing range of the pendulum shaft.

Similar with the case of $\alpha=0^\circ$, first, the variation pattern of the interpolation error is determined, and then the meridional profile curve of the processed ellipsoid is respectively divided into 1, 2 and 100 interpolation segments for analysis, with the simulated result as shown in Figure 8.

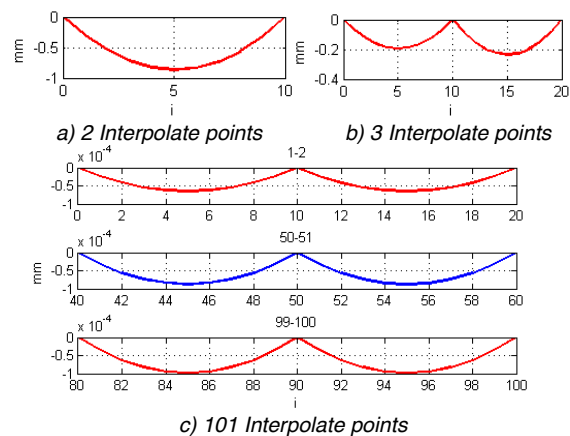


FIGURE 8. The simulated result ($\alpha = 31.2544^\circ$)

The simulated result is similar to the case of parallel spindles, except that the maximum value of the interpolation error is found in the last interpolation segment. This is related to the angle α between the two spindles, the value of which can greatly affect the position of the

maximum interpolation error segment, as the maximum value of interpolation error moves towards the *R* end with the increase in the angle α . During processing, if the surface shape error is required as $PV < 1 \mu\text{m}$, then at the angle $\alpha = 31.2544^\circ$ between the two spindles, the calculated number of the rough interpolation points is 33, and the maximum radial error is found in the 32nd interpolation segment, with a value of $-0.961 \mu\text{m}$.

CONCLUSION

When the angle α between the two spindles increases, the swing angle of pendulum shaft can be increased for the same interpolation segment, which helps the existing resolution of the lathe to meet the processed requirements. For the workpiece of the same length, the range of motion of the workpiece in the *z* direction can be decreased, but a large α is not necessarily optimal for processing. For this reason, selecting an appropriate angle α will help improve the processing quality. After several simulation tests, it is appropriate to select $\alpha = 21.307^\circ - 21.3246^\circ$. When $\alpha = 21.307^\circ$, corresponding to 14 interpolation points, the maximum error is found in the first interpolation segment, with a value of $0.973 \mu\text{m}$; when $\alpha = 21.3246^\circ$, corresponding to 14 interpolation points, the maximum error is found in the 13th interpolation segments, with a value of $0.975 \mu\text{m}$. Of course, a proper increase in the number of interpolation points is conducive to improving the quality of the processed workpiece. Meanwhile, the swing velocity ω of rotation axis is lowered, to reduce the linear velocity of the workpiece, thereby helping to improve the surface roughness of the processed workpiece.

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COMPARISON OF ANALYTICAL MILLING STABILITY ANALYSES WITH TIME DOMAIN SIMULATION

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INTRODUCTION

In the past few decades, machine and cutting tool technology has increased in both capability and complexity. Despite the significant advancements, however, self-excited vibrations (or chatter) remain a limiting factor for material removal rates and part quality.

In 1965, Tobias showed that chatter is self-excited vibration which results from regeneration effects on instantaneous chip thickness [1]. The cutting edge removes a chip that was produced by in the previous pass (the prior revolution in turning or tooth in milling). The chip thickness, which affects the force and therefore the vibration response, depends on the phase between the previously cut surface and the current vibration. This understanding led to the development of analytical algorithms which are used to generate a stability map of the limiting axial depth of cut, b_{lim} , versus spindle speed, Ω . This map is known as a stability lobe diagram. As an alternative, the governing equations of motion may be solved in the time domain to define the machining behavior (time domain simulation).

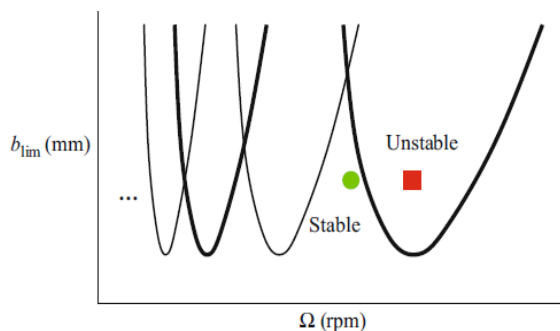


Figure 1. Stability lobe diagram [2].

In this paper, two popular frequency domain stability analyses are compared to time domain simulation results. A new stability metric is introduced to automatically identify the time domain simulation stability.

FREQUENCY DOMAIN STABILITY ANALYSIS

Frequency domain stability analysis is used to determine the global stability of a milling operation as a function of spindle speed and depth of cut. Two popular frequency domain techniques are:

1. average tooth angle approach developed by Tlusty [3]
2. Fourier series approach developed by Altintas and Budak [4].

The average tooth angle approach assumes an average tooth angle in the cut (the mean of the entry and exit angles) and, therefore, an average force direction. This assumption creates an autonomous, or a time-invariant, system. Tlusty then made the use of directional orientation factors to first project this average force into the x (feed) and y-directions and, second, project these results onto the surface normal.

The Fourier series approach transforms the dynamic milling equations into a time-invariant, but radial immersion-dependent, system. The frequency domain algorithm accounts for the x and y-direction tool deflections and uses a truncated Fourier series (mean value only) to represent the cutting force.

TIME DOMAIN STABILITY ANALYSIS

Time domain simulation is used to calculate the forces and displacements during milling. It gives local, rather than global, information about the process behavior for a given spindle speed and axial depth of cut. The simulation implemented in this study [2] is based on the "Regenerative Force, Dynamic Deflection" model described by Smith and Tlusty [5]. The simulation proceeds as follows.

1. The instantaneous chip thickness is determined using the vibration of the previous and current teeth at the selected tooth angle.

2. The cutting force components are calculated as the product of the chip width (axial depth), chip thickness, and specific force coefficients.
3. The force is used to find the new displacement by Euler (numerical) integration of the second order, time delay equations of motion.
4. The tooth angle is incremented and the process is repeated.

The simulation takes into account the non-linearity that can occur if the tooth leaves the cut due to large vibration amplitude. It can also accommodate cutter teeth runout, variable teeth spacing, and variable helix angles.

To automatically identify stable or unstable behavior from the simulated displacements, a new stability metric was defined. The new metric

is: $M = \sum_{n=1}^N \left| \frac{(x_{ts}(i) - x_{ts}(i-1))}{N} \right|$, where where x_{ts} is the vector of once-per-tooth sampled tool displacements in the x-direction and N is the number of points. Similarly, this metric can also be applied to the y-direction tool displacements or the workpiece displacements. The absolute value of the difference in the successive sampled points is summed. If the cut is stable (forced vibrations), the sampled points repeat and the M value is nominally zero. In this research, $M < 1 \mu\text{m}$ was used to identify a stable cut, while an M value greater than or equal to $1 \mu\text{m}$ indicated an unstable cut.

RESULTS

To compare the frequency and time domain analyses, the system dynamics were selected and the stability was identified over a range of spindle speeds and axial depths of cut. As an example, the system dynamics for a single degree of freedom (DOF) system were specified as shown in Table 1.

Table 1. Single DOF dynamics, flexible tool.

Tool dynamics	x	y
Natl. freq. (Hz)	750	750
Damping (%)	5	5
Stiffness (N/m)	5×10^6	5×10^6
Workpiece dynamics	x	y
Natl. freq. (Hz)	750	750
Damping (%)	5	5
Stiffness (N/m)	5×10^{10}	5×10^{10}

The diameter of the two-flute milling cutter was 20 mm in all cases. The specific cutting force for the aluminum alloy workpiece was $700 \times 10^6 \text{ N/m}^2$ and the force angle was 68 deg [2].

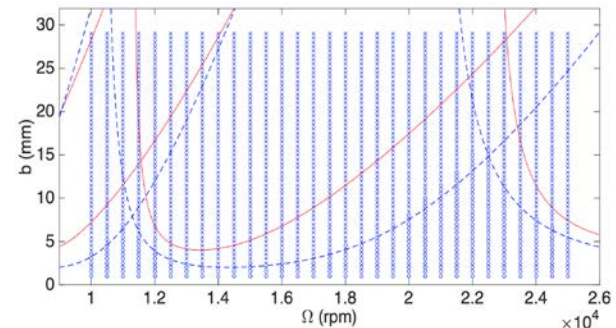
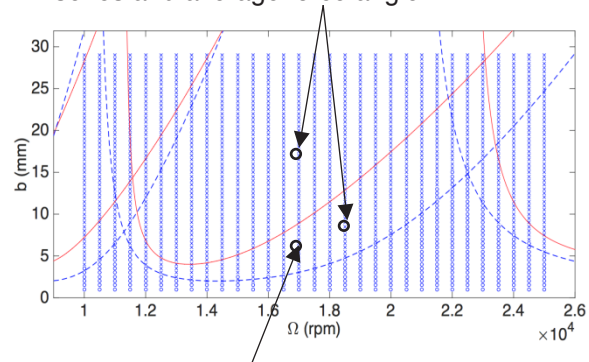


Figure 2. Comparison of frequency and time domain stability results.

Figure 2 shows the two frequency domain solutions superimposed over a grid of time domain simulation results, where the solid line represents the average tooth angle approach, the dotted line represents the Fourier series approach, a circle indicates a stable time domain simulation result, and an $\times$ identifies an unstable result. The up milling radial immersion used to generate this figure was 50%.

Stable points (o) are correct for Fourier series and average force angle.



Unstable point ($\times$) is correct for Fourier series, but incorrect for average force angle.

Figure 3. Procedure for counting correct and incorrect grid points to determine accuracy of frequency domain stability analyses.

To quantify the accuracy of the frequency domain stability analyses, the number of incorrect points was tallied for both approaches and the percent correct score was calculated. For each point in the simulation grid, the stability

was determined (time domain simulation was assumed to be correct). If an unstable point was below the frequency domain stability limit, the point was considered incorrect. Similarly, if a stable point occurred above the frequency domain limit, it was identified as incorrect. Figure 3 demonstrates the counting procedure graphically. This figure has 1767 time domain simulation points in the grid. There are 378 incorrect points for the average tooth angle approach, which corresponds to a percent correct score of 78.6%. The Fourier series approach has 25 incorrect points, which gives a 98.6% correct score.

This procedure was repeated for multiple cases, including both up milling and down milling, different radial immersions, and varying system dynamics:

1. single DOF symmetric tool (Table 1)
2. single DOF asymmetric tool
3. two DOF symmetric tool
4. two DOF asymmetric tool
5. single DOF symmetric (equal tool-workpiece)
6. single DOF symmetric (unequal tool-workpiece).

A special low damping case for single DOF tool dynamics was also considered. To compare the two frequency domain analyses, the average percent correct score for all six dynamic cases is provided in Tables 2 and 3 as a function of radial immersion (100% is slotting) for both up and down milling.

Table 2. Comparison of up milling percent correct scores from all six dynamic cases.

Radial immersion	Frequency domain analysis	Average score (%)
100%	Fourier series	96.8
	Average tooth angle	77.0
75%	Fourier series	96.6
	Average tooth angle	77.8
50%	Fourier series	96.9
	Average tooth angle	82.1
25%	Fourier series	95.8
	Average tooth angle	91.1
5%	Fourier series	93.9
	Average tooth angle	90.5

Table 3. Comparison of down milling percent correct scores from all six dynamic cases.

Radial immersion	Frequency domain analysis	Average score (%)
75%	Fourier series	95.2
	Average tooth angle	90.4
50%	Fourier series	97.3
	Average tooth angle	82.3
25%	Fourier series	95.3
	Average tooth angle	90.3
5%	Fourier series	94.3
	Average tooth angle	91.5

Tables 2 and 3 summarize the average percent correct score for the six system dynamics cases at each radial immersion. It is observed that the Fourier series approach more closely agrees with the time domain simulation results. The percent correct score for the average tooth angle approach tends to increase as the radial immersion decreases.

SINGLE DOF LOW DAMPING

A special case was also simulated where the tool was lowly damped and much more flexible than the workpiece. The system dynamics are summarized in Table 4. Time domain simulations were performed with low radial immersion (5%) for both up milling and down milling. The results are presented in Figs. 4 and 5. It is observed that neither frequency domain approach accounts for the new unstable zone between 14000 and 16000 rpm (period-2 bifurcation). This occurs due to the low damping and low radial immersion (highly interrupted cutting); see Table 5.

Table 4. Single DOF low damping dynamics.

Tool dynamics	x	y
Natl. freq. (Hz)	750	750
Damping (%)	0.5	0.5
Stiffness (N/m)	5×10^6	5×10^6
Workpiece dynamics	x	y
Natl. freq. (Hz)	500	500
Damping (%)	5	5
Stiffness (N/m)	5×10^{10}	5×10^{10}

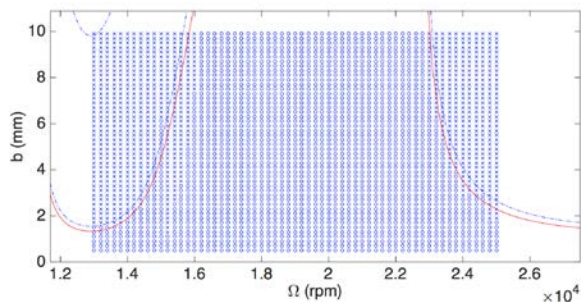


Figure 4. Single DOF low damping case (5% up milling).

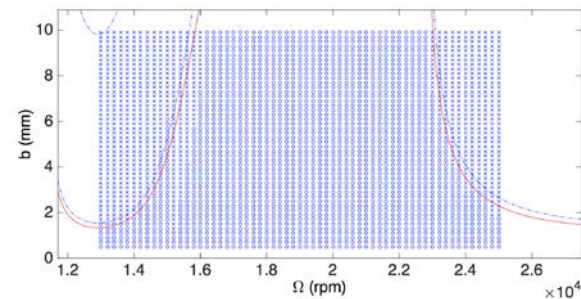


Figure 5. Single DOF low damping case (5% down milling).

Table 5. Single DOF low damping results.

Radial Immersion	Frequency domain analysis	Percent score
5% up milling	Fourier series	93.4
	Average tooth angle	94.8
5% down milling	Fourier series	93.8
	Average tooth angle	95.0

VARIATION OF M METRIC WITH AXIAL DEPTH OF CUT

As defined previously, the M metric defines stability using the difference between successively sampled points. As seen in Fig. 2, the stable (o) and unstable (x) points were identified using a threshold M value of $1 \mu\text{m}$ ($M < 1 \mu\text{m}$ for stable and $M \geq 1 \mu\text{m}$ for unstable). This section describes how the metric value changes with an increase in axial depth of cut. The variation in the M values with axial depth of cut and the corresponding change in behavior (stable vs. unstable) is displayed in Fig. 6. Results for multiple spindle speeds are included. It is seen that the M value changes rapidly at the onset of instability.

For example, at a spindle speed of 11500 rpm, the M value for the first unstable point is $7 \mu\text{m}$. For 15500 rpm, it is $6 \mu\text{m}$. For 22000 rpm, it is $17 \mu\text{m}$ and it is $26 \mu\text{m}$ for 25000 rpm. This

validates the $1 \mu\text{m}$ threshold applied in this study.

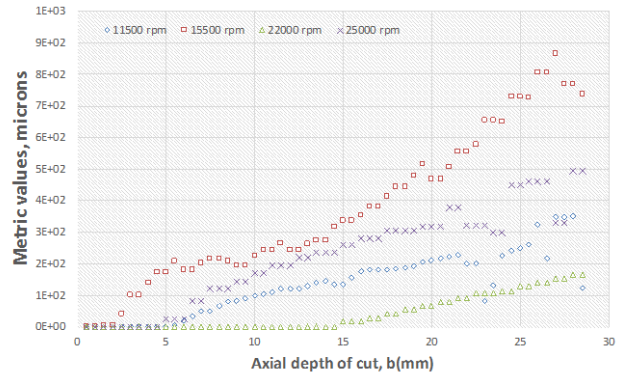


Figure 6. Variation in M with axial depth.

CONCLUSIONS

This paper compared two popular frequency domain milling stability solutions with time domain simulation. A new metric was provided to automatically identify the stability of the time domain simulation results. It was found that the Fourier series analysis more closely matched time domain stability predictions than the average tooth angle approach.

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VOLUMETRIC ERROR ANALYSIS OF THE TRICEPT PKM BASED 5-AXIS MACHINE TOOL CONSIDERING THE GEOMETRIC ERRORS OF LINEAR AXES

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INTRODUCTION

Parallel kinematic machines (PKMs) such as Stewart platform and delta robot are widely used in industrial field due to its high stiffness and agile movement. During the last decade, Tricept PKM based five-axis machines with hybrid (serial-parallel) kinematics have been developing in machine tool industry. XT models of Exechon enterprised LLC, which is representative commercial model using the Tricept PKM concept, have wide working space compare to other PKMs.

Despite these advantages, the accuracy of PKMs is still lower than conventional serial type machine tools due to numerous error sources such as assembly error between joints, thermal displacement error and deformation by gravity. [1] To improve the positioning accuracy, error definition, modeling and compensation strategy of the Stewart platform or Delta robot based machines were attempted. [2,3] Inverse/forward kinematics models of the Exechon PKM have been proposed by Bi et al. [4] Because the Exechon PKM is mainly applied to machine tool, the compensation of the volumetric error should be conducted to improve the form accuracy of product. However, the research about the analysis and compensation method is insufficient.

In this research, the analysis of the volumetric error considering the linear displacement error of the Exechon PKM type machine tool is presented. To calculate the volumetric error according to the error sources such as pitch error of a ball-screw and thermal displacement error, working space is calculated with the hybrid kinematics model. Then, the volumetric error are calculated by applying these error sources which affects to linear displacement error.

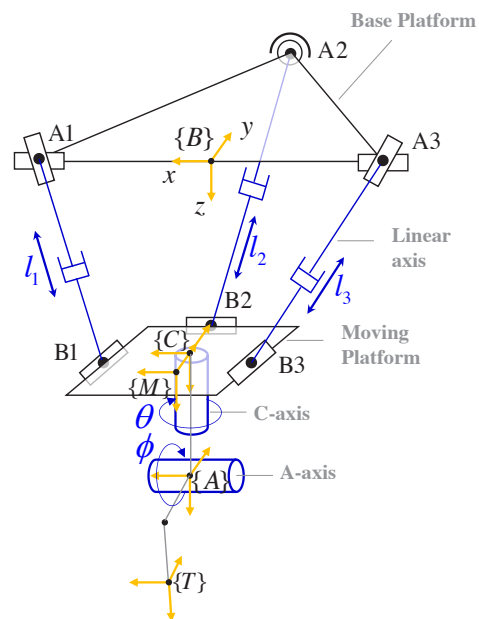


FIGURE 1. Structural Configuration of the Tricept PKM based 5-axis Machine Tool

WORKING SPACE CALCULATION OF THE TRICEPT PKM BASED MACHINE TOOL

Structural Configuration

Tricept PKM based 5-axis machine tool, namely Exechon PKM (EPKM), consists of base platform, moving platform, two rotary axes (A, C-axis), and tool, such as Figure 1. The base platform and the moving platform are connected to three limbs including the linear driving axes in parallel. The joints (A1, A3) between 1st-, 3rd-limb with the base platform are universal joints. 2nd-limb is connected to one universal joint and one revolute joint (similar to spherical joint, A2). The three limbs are also connected to the moving platform by revolute joints (B1, B2, B3),

respectively. On the other hand, tool (or spindle) is serially linked with the C- and A-axis in series.

Working Space Calculation using Forward Kinematics

The mathematical models of both inverse kinematics and forward kinematics are difficult to obtain the analytic solution because the machine structural configuration is combined serial and parallel mechanism. Therefore, the solutions of inverse kinematics and forward kinematics should be calculated using numerical analysis methods. The forward kinematics of the EPKM determine the position $\mathbf{p}_t = [p_{tx} \ p_{ty} \ p_{tz}]$ and orientation $\mathbf{t} = [t_x \ t_y \ t_z]$ of the tool using the length l_1, l_2, l_3 of three limbs and rotation angles α, γ of two rotary axes. First, the origin and orientation of the moving platform coordinate system (MCS) relative to base coordinate system (BCS) according to the length of the limbs are defined from the design values and constraint relations between BCS, MCS and joint coordinate systems, as follows.

$$\begin{aligned} l_1 &= \text{Sqrt}[(p_{Mx} - a)^2 + p_{My}^2 + p_{Mz}^2 - 2abu_x \\ &\quad + 2bcu_y + b^2] \\ l_2 &= \text{Sqrt}[p_{Mx}^2 + (p_{My} - c)^2 + p_{Mz}^2 - 2cdv_y + d^2] \\ l_3 &= \text{Sqrt}[(p_{Mx} + a)^2 + p_{My}^2 + p_{Mz}^2 - 2abu_x \\ &\quad - 2bcu_y + b^2] \end{aligned} \quad (1)$$

where, a and c mean design values (constant) of joint position relative to BCS. And, b and d present design values (constant) of joint position relative to MCS. The origin of MCS relative to BCS is $\mathbf{p}_M = [p_{Mx} \ p_{My} \ p_{Mz}]$. u_x, u_y and v_y are entities of the orientation matrix of MCS relative to BCS, and are function of $\mathbf{p}_M$. Therefore, the origin and orientation of the MCS are obtained by using the numerical method such as Newton-Raphson method.

TABLE 1. Minimum and maximum length and angle (software limit) of driving components

Components	Limits of length and angle	
	Minimum	Maximum
1 st Limb	893.0 mm	1593.0 mm
2 nd Limb	935.0 mm	1635.0 mm
3 rd Limb	893.0 mm	1593.0 mm
C-axis	-360°	360°
A-axis	-180°	5°

Because the moving platform and rotary axes are connected in series, the tool position and orientation easily calculated.

$$\begin{aligned} \mathbf{p}_T &= \mathbf{p}_M + {}^M\mathbf{p}_A + {}^A\mathbf{p}_T \\ &= \mathbf{p}_M + R_M^B [0 \ e \ f]^T + R_S^B [0 \ -g \ h]^T \end{aligned} \quad (2)$$

where, ${}^M\mathbf{p}_A$ is the origin of A-axis coordinate system relative to MCS, and ${}^A\mathbf{p}_T$ is the position of the tool relative to A-axis coordinate system. e, f, g and h are design values (constant) from the MCS to tool coordinate system. R_M^B and R_S^B are rotation matrix of the MCS and S-coordinate system, which is a virtual coordinate system, relative to BCS, respectively.

The working space of one of the EPKM model (XT700S) is obtained by calculating the presented forward kinematics model as shown in Figure 2. The limits of length of limbs and angles of rotary axes are given such as Table 1.

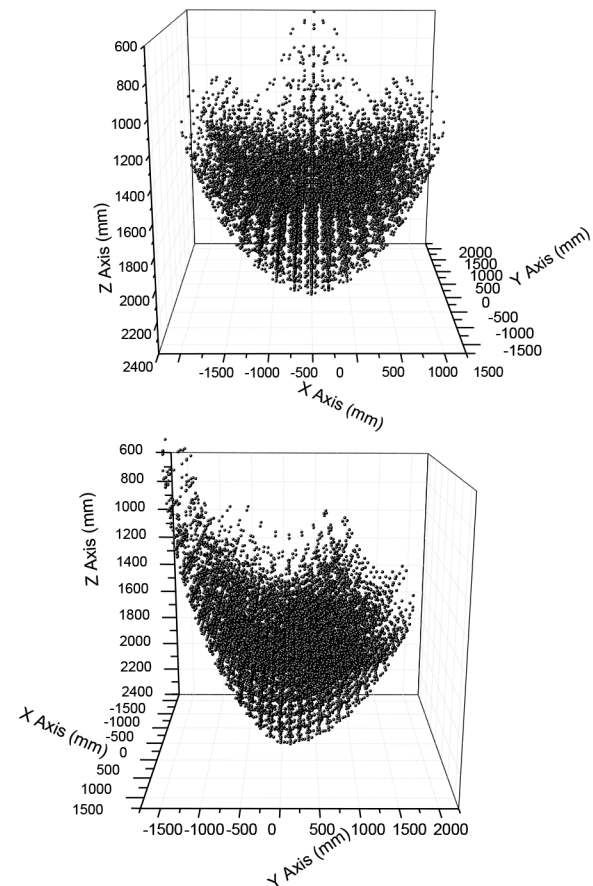


FIGURE 2. (Top) front view (bottom) and side view of the working space of the EPKM considering the limits

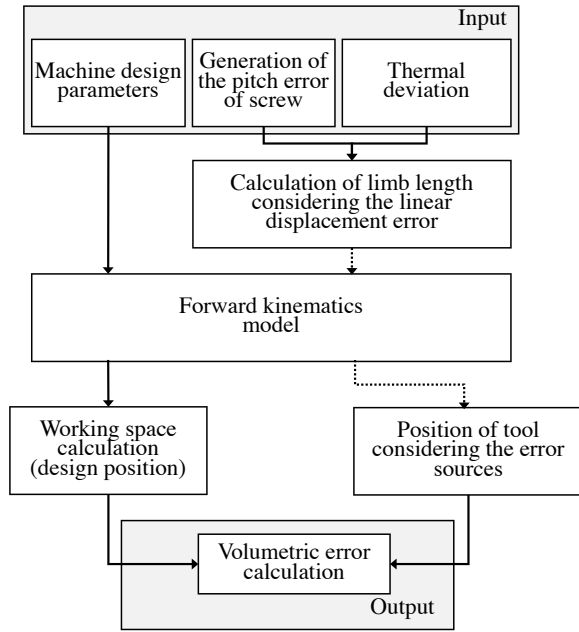


FIGURE 3. Simulation procedure for calculating the volumetric error of EPKM according to pitch error and thermal displacement error of the linear axes

Shapes of the working space of Tricept based PKMs are almost a 'Shield'.

VOLUMERIC ERROR ANALYSIS CONSIDERING THE LINEAR DISPLACEMENT ERROR OF LINEAR AXES

Procedure for Volumetric Error Calculation

In case of the conventional 3-axis machine tools, linear displacement error of the driving axis affects to various error sources such as pitch error of screw and thermal displacement error. The positioning accuracy is significant influenced by the linear displacement error compare to other geometric errors. The linear displacement error is also important error component in the EPKM because the length of limb which includes the linear axis is relatively longer than other PKMs.

Calculation procedure of the volumetric error such as Figure 3 is basically used the forward kinematics shown in previous section. Although EPKM has the numerous error components, it is assumed that there is no errors except the linear displacement errors Δh , Δk , Δl .

$$l'_1 = l_1 + \Delta l_1, l'_2 = l_2 + \Delta l_2, l'_3 = l_3 + \Delta l_3 \quad (3)$$

where, l_1, l_2, l_3 are the lengths of limbs considering the linear displacement errors.

The volumetric error ΔW is obtained by subtracting the position of tool $\mathbf{p}_T$ considering the error sources from design position $\mathbf{p}_T$.

$$\Delta W = [\Delta x \quad \Delta y \quad \Delta z] = \mathbf{p}'_T - \mathbf{p}_T \quad (4)$$

Simulation

To confirm effects of the pitch error and thermal displacement error in the volumetric error, computer simulations with the presented procedure are conducted. The linear displacement errors, which are influenced by the pitch errors of the three linear axes are generated arbitrary such as Figure 4. The volumetric error according to the pitch errors is obtained as shown in Figure 5(a). The linear displacement errors which are affected by the thermal displacement error as well as the pitch errors are generated according to the thermal variation. The volumetric errors are calculated as shown in Figure 5(b), (c). Results of three cases are summarized in Table2.

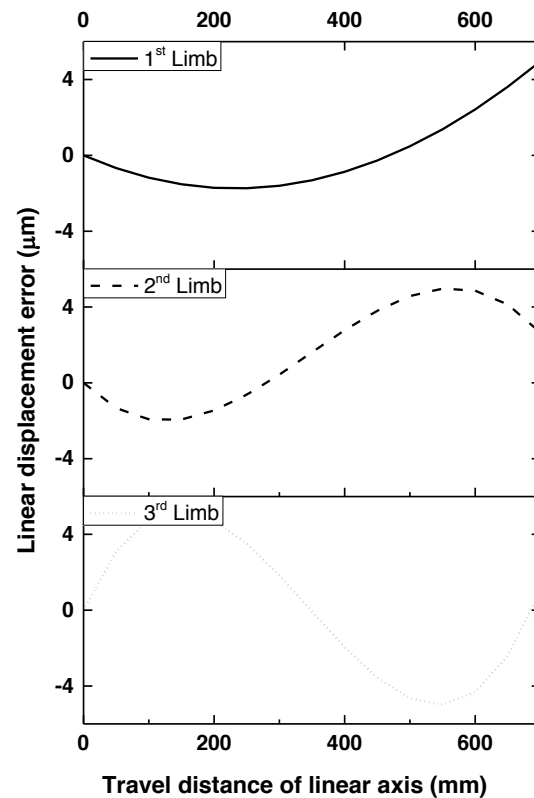


FIGURE 4. Generated linear displacement errors of each linear axes (limbs) considering the pitch errors for computer simulation

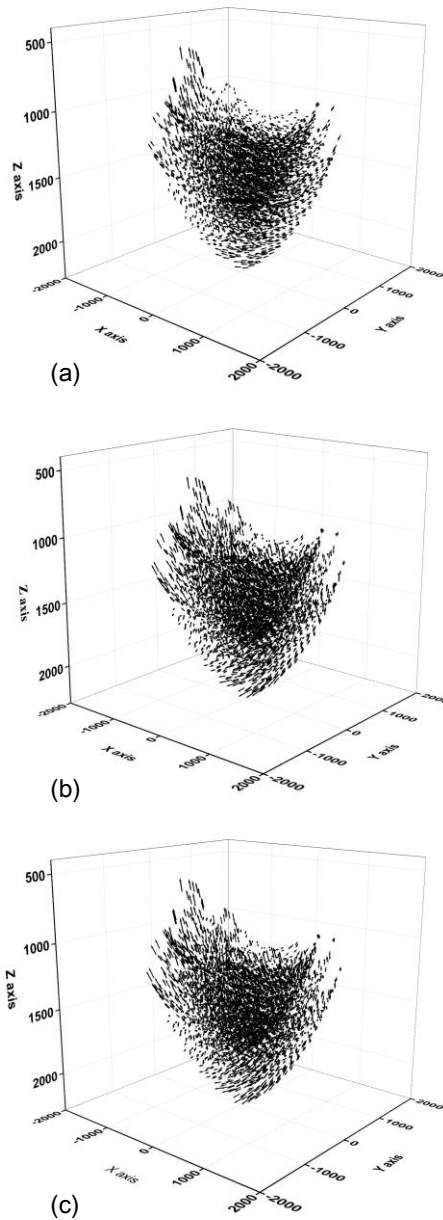


FIGURE 5. Volumetric error according to (a) only pitch errors (b) pitch errors and thermal displacement errors (1°C increment) (c) pitch error and thermal displacement error (2°C increment)

The Measurement of Linear Displacement Error using Laser Tracking System

Due to the alignment and restriction of measurement path, a laser interferometer system and a double ball-bar are not suitable to measure the EPKM. Therefore, a laser tracking system has been studied as an alternative to conventional measurement systems, it is possible to easily measure the linear displacement error of linear axis of the EPKM.

CONCLUSION

In this study, the techniques for working space calculation and volumetric error evaluation using the mathematical model of forward kinematics are introduced. The volumetric error is significantly affected by the linear displacement errors of the linear axes in the EPKM with the wide working space. Linear displacement error can be measured by using the laser tracker system, and can be compensated by adjusting the pitch error parameter in commercial controller. Also, to improve the positioning accuracy of the EPKM, real-time monitoring for the temperature of each limbs is needed.

ACKNOWLEDGEMENTS

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TABLE 2. Maximum magnitude of the volumetric error at each simulation result

Case	Parameter	Value
Only pitch error	magnitude	25.7 μm
	Δx	-6.7 μm
	Δy	-23.5 μm
	Δz	-5.8 μm
Pitch error and thermal displacement error (1°C increment)	magnitude	41.2 μm
	Δx	-7.6 μm
	Δy	-34.7 μm
	Δz	-20.9 μm
Pitch error and thermal displacement error (2°C increment)	magnitude	65.1 μm
	Δx	25.8 μm
	Δy	-47.7 μm
	Δz	-36.0 μm

THE HETEROGENEITY OF MATERIALS IN THE ULTRA-PRECISION CUTTING BASED ON THE PROBABILISTIC FINITE ELEMENT METHOD

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INSTRUCTIONS

Ultra-precision cutting is significant for mold manufacture of optical element. The mechanism investigation of ultra-precision cutting has always been the hotspots. As the real time online observation experiment is nearly impossible to be conducted at the time of cutting process, it is difficult to reveal the mechanism of ultra-precision cutting directly. With the development of finite element method, the technique to simulate the cutting process is widely used. Finite element analysis (FEA) allows for a better understanding of cutting process in detail. The process of material removal can be simulated and the causes thereof can be revealed. Schermann et al. calculated cutting forces by using extensive mass scaling and an explicit time integration scheme to reduce the computation time [1]. Mamalis et al. conducted the thermal model of surface grinding [2]. Furthermore, Lohkamp et al. used implicit time integration to research titanium with the consideration of crystal plasticity [3]. Aurich et al., simulated the burr and chip formation in 3D finite element model [4-5]. However, in the ultra-precision cutting, the rate of material removal is fairly low to obtain the excellent surface. No studies mentioned above concerned the heterogeneity of materials in micro level, which is fairly important for the ultra-precision cutting.

THEORY BACKGROUND

In the cutting process, both elastic and plastic deformation will occur. According to the theory of metallic elastic and plastic deformation, the incremental stress-strain model is proper to express this process. Full stress-strain model is usually used to describe the process of monotonic loading. For the process of plastic and elastic deformation which contains many loading and unloading, the incremental stress-strain model is needed.

Firstly, in the process of elastic deformation, the Hook theory is still available. Here to convert it to an incremental format:

$$d\sigma = E d\varepsilon \quad (1)$$

When the load keeps increasing after the yield point is reached, the material will come into the stage of plastic deformation. Usually in different direction there are different yield stress and they make up a closed surface in the two-dimensional coordinate system, which is called yield surface and yield function. In the process of loading and unloading, the yield surface will move with the load history. In real process of cycled load the isotropic and kinematic hardening are existed in the same time, which is called mixed hardening. It can be expressed as:

$$f(\sigma, \alpha, k) = [\sigma - (1 - M)\alpha]^2 - [(1 - M)\sigma_0 + Mk]^2 \quad (2)$$

Where M is a parameter represents the mixed degree and $0 \leq M \leq 1$.

The increment of hardening parameter in every cycle can be expressed as:

$$dk = h d\lambda \quad (3)$$

Where h is a scalar quantity function.

When the stress is located in the boundary between elastic area and plastic area, the elastic and plastic relationship should be met in the same time, which can be expressed as:

$$\frac{\partial f}{\partial \sigma} d\sigma + \frac{\partial f}{\partial k} dk = 0 \quad (4)$$

In the process of plastic deformation, to imitate the flow rule in the hydromechanics, the relationship between stress and strain is expressed as:

$$d\varepsilon^p = d\lambda \frac{\partial f}{\partial \sigma} \quad (5)$$

Where f is the yield function and in this paper the mixed hardening is adopted.

After the definition of yield stress, hardening and flow rules, the constitutive model in the process of plastic deformation will be derivated. In the process of plastic deformation, the deformation usually consists of elastic deformation and plastic deformation:

$$d\varepsilon = d\varepsilon^e + d\varepsilon^p \quad (6)$$

In the real load process, the plastic deformation remains after the load is removed. As a result the plastic deformation increments $d\varepsilon$ can be omitted here:

$$d\sigma = E d\varepsilon^e \quad (7)$$

Make a calculation among the function above:

$$d\varepsilon = \frac{d\sigma}{E} + d\varepsilon^p = \frac{1}{E_t} d\sigma \quad (8)$$

Where E_t is equivalent Young's modulus which is expressed as:

$$\frac{1}{E_t} = \frac{1}{E} \left(\frac{\partial f}{\partial \sigma} \right)^2 E - \frac{1}{E_t} \frac{d\sigma}{\frac{\partial f}{\partial k} h} \quad (9)$$

So far an incremental stress-strain model which expresses the deformation under the recycled load is founded.

MODEL DEVELOPMENT

Under microscopic observation common metallic materials tends to show a significant distribution of grains and grain boundaries. The shape of grain is irregular and the scale of strain is not uniform. Considering the geometric properties of the process of liquid metal cooling, nucleation and crystallization, the Voronoi picture can be used to describe the microstructure of the metallic materials. As is shown in Fig.1, the metallic microstructure founded by Voronoi is very similar in shape and dimensions to the real microstructure of common metallic materials.

In this paper the method of mixed programming between Matlab and Python is used. In the matlab environment Voronoi picture is founded and its geometric information is output. In the Python environment built in Abaqus compiler the geometric information is received from the previous step and its reconstruction is finished. Implementation process is as follows:

- To establish a two-dimensional Voronoi diagram and output its geometric information in the matlab environment. In the matlab

environment MPT toolbox is called to generate two-dimensional and three-dimensional Voronoi pictures. Location and topology information of points and segments for 2-D Voronoi can be output directly. For 3-D voronoi location and topology information of points, lines and surface can be generated after the process of a specific algorithm. Then to output and save the information as an Excel file format.

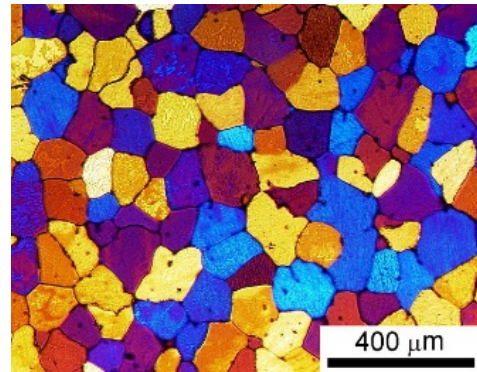


FIGURE 1. Real microstructure of common metallic materials.

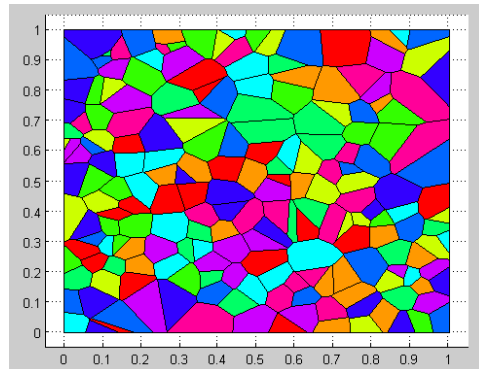


FIGURE 2. Metallic microstructure founded by Voronoi.

- To read and reconstruct the geometric information of materials in Abaqus environment. There is a built-in PDE interface in Abaqus which can be used to run the Python language to control the process of simulation by rpy file, including the process of modeling, loading, and other post-processing analysis step. There is also an XLRD module built in the Python language which can read the geometric information on the previous step in the format of Excel and rebuild it in the Abaqus

environment so that the material microstructure characteristics required can be built in Abaqus environment.

- After running the program written according to the flow diagram in Matlab and Python a model which consists of a series of irregular polygon and bodies will be shown in Abaqus. In this model each small part can be seen approximately as an independent grain correspond to the real grain in common metallic materials. Apparently this new model is more similar to the real microstructure of common metallic materials compared with the traditional finite element model. These two-dimensional and three-dimensional microstructure built in Abaqus pre-processing environment. Comparing the microscopic picture of common metallic materials and new metallic microstructure established in this paper, it is concluded that these two parts share a high degree of similarity. In order to verify the accuracy of the model, the finite element model of the new microstructure is used to simulate the material properties and the results of simulation will be compared with the experimental results.

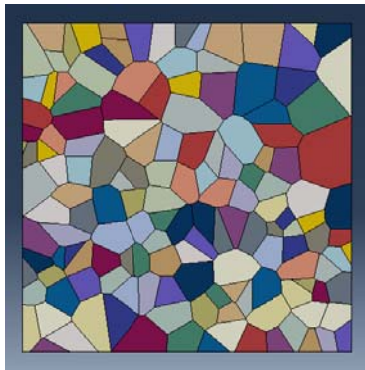


FIGURE 3. 2-D diagram of the new metallic microstructure in ABAQUS.

- To perform a stochastic distribution of material properties in the new model. In the microstructure model, each small part can be seen approximately as an independent grain correspond to the real grain in common metallic materials. According to the approach of stochastic distribution, each grain in the material has a different mechanical property which performs the phenomenon that the yield point of each grain is different from each other. By nano-

indentation experiments the theory of distribution is obtained and in the next step this theory will be introduced into the microstructure.

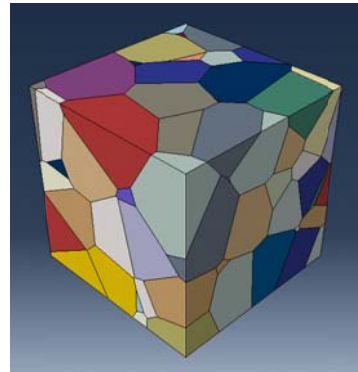


FIGURE 4. 3-D model of the new metallic microstructure in ABAQUS.

This part is fulfilled by Python program in the FEA software Abaqus. Firstly a series of random material properties is obtained by some module. Secondly a series of unit number is obtained by another module and the unit number is coordinated with the material property. Lastly the property will be matched with the unit one by one until all grains has been assembled with a property value.

As is shown in Fig. 5, after the stochastic distribution the properties of all grains in the material show a theory of Gaussian distribution.

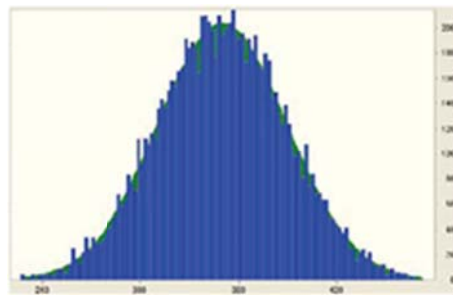


FIGURE 5. Stochastic distribution the properties of all grains in the material.

SIMULATION

After the material model is founded, FEA analysis could be conducted on the cutting simulation model. Compared with the traditional modeling methods, the new model can reflect

the microscopic structure of the material better under load force and deformation. To reduce the amount of work and to simplify the calculation process, two-dimensional microstructure is used to conduct the cutting simulation. In the simulation process, in accordance with the periodic boundary conditions, some areas of sample are picked up with load on the sample upper surface. The simulation model is shown below, some parameters of the material part are set as follows:

TABLE 1. Scale parameters for materials in simulation

Sample size	1mm * 1mm
Number of grains	400
Average grain scale	50um
Parameters	Values
Density	4.5g/cm ³
Young's modulus	110GPa
Poisson's ratio	0.34
Yield limit	860MPa
Tensile strength	967MPa
Density	4.5g/cm ³

CONCLUSION

With the introduction of Voronoi picture in the modeling process to describe the material microstructure and stochastic distribution of material properties, the new microstructural finite element model is able to restore the microstructure of the material better. Obviously, the model established in this letter could be more accurate to reflect the material behavior in the ultra-cutting process. In the future, the new model will be compared with the simulation model using uniform property materials. The simulation results also need to be verified by experiments.

ACKNOWLEDGEMENT

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THE LEAST INCREMENT TEST OF MACHINE TOOLS AND THE INFLUENCE FACTORS

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INTRODUCTIONS

This paper described a test method for machine tools performance. This test can quantify the machine's ability to make small corrections in position^[1]. It could not only estimate position accuracy and repeatability by determining the least increment of machine tools, but also rapidly obtain the reversal errors of machine tools. Due to the performance of machine tools is effected by machine construction and control parameters, the least increment test could be used to research the effects of these factors. In this paper, the least test is applied on a linear axe with closed loop control to demonstrate effectiveness and feasibility of the proposed method.

TEST EQUIPMENT

The setup for such test is a inductance micrometer having excellent precision. Its resolution is as high as 0.01 μ m, as shown in *FIGURE 1*.

A computer is necessary to gather and analysis data.

METHOD FOR MEASUREMENT AND ANALYSIS

The inductance micrometer is fixed on the stationary along the movement direction of machine tools to assure the displacement of machine tools can be measured and recorded.

The machine moves the ten positive increments, and then ten negative increments. Each step stays for a while to wait for the machine stable. The increment shall start at the resolution of the machine and shall be increased until each step of the machine is clearly measured and recorded. *FIGURE 2*. shows the sample results from the small increment test.

The least increment test could obtain two parameters of machine tools, the least increment and the reversal error. The least increment is the commanded increment at which the ten positive

steps and the negative steps can be counted clearly and the overshoot of each step does not exceed 5% of the step. Reversal error is the value of the first negative step minus the commanded increment.



FIGURE 1. Inductance micrometer, resolution 0.01 μ m

FIGURE 3. shows the displacement of machine for a step. Maximum value of the step is $X_{\max}$, Stable value of the step is X_s , and overshoot of each step is $\delta\%$.

$$\delta\% = \frac{(X_{\max} - X_s)}{X_s} \times 100\%$$

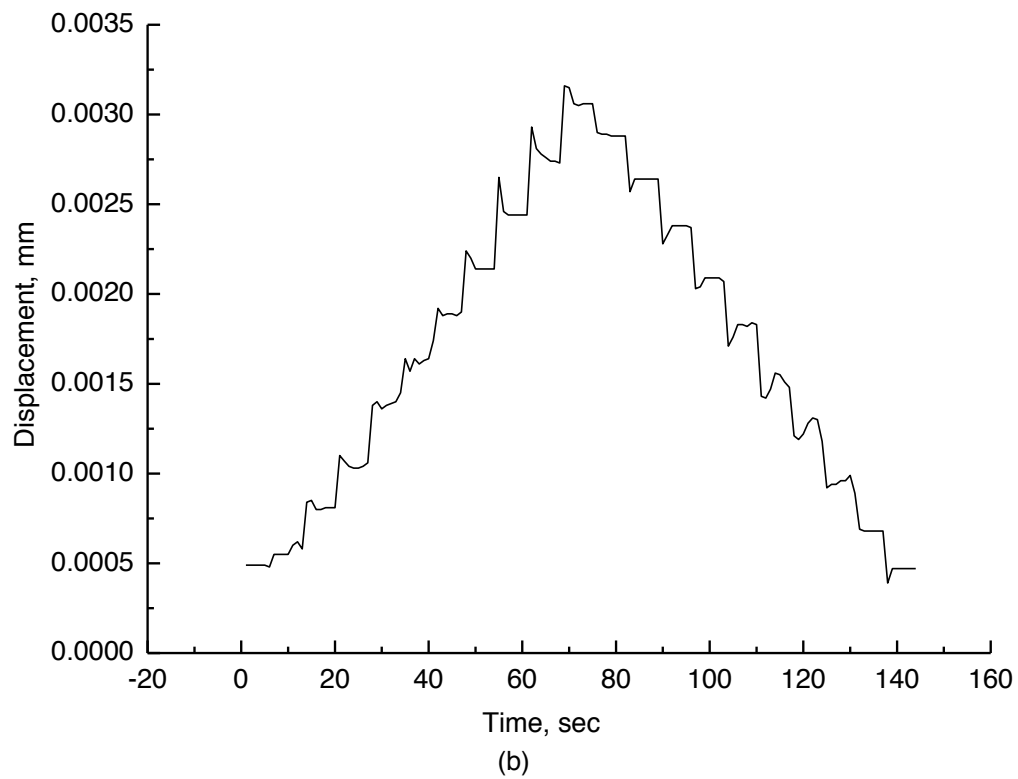
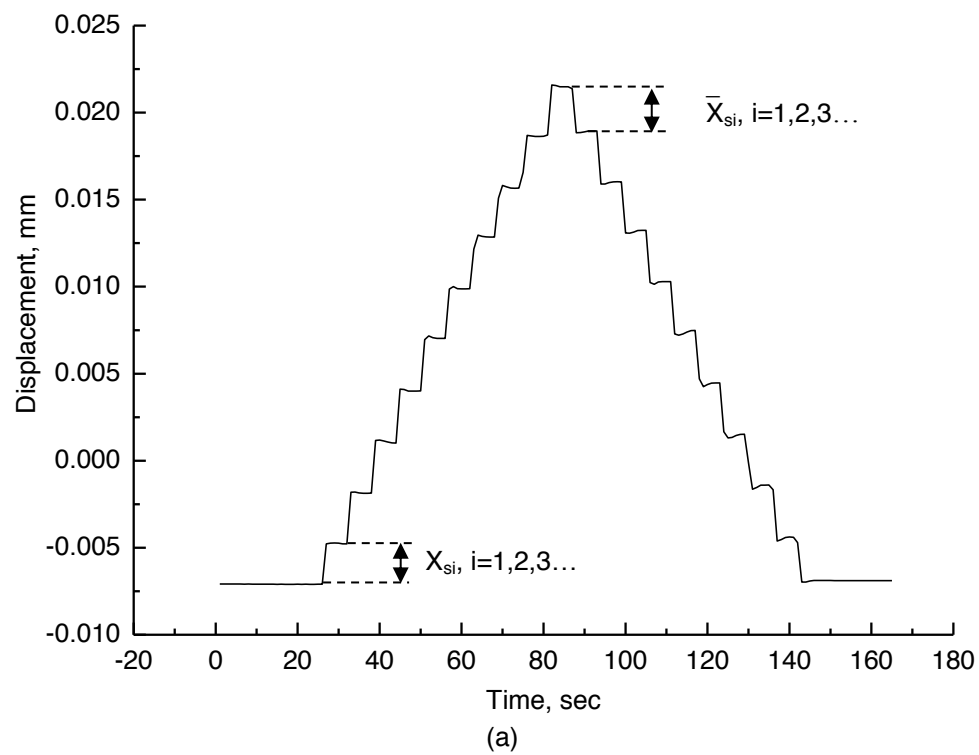


FIGURE 2. Sample results from the small increment test

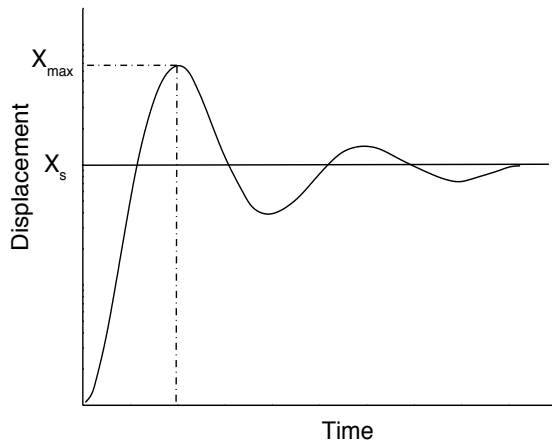


FIGURE 3. The displacement for a step

Commanded increment is represented by X . Stable value of each positive step is represented by X_{si} ($i=1,2,3,\dots$). Stable value of each negative step is represented by $\bar{X}_{si}$ ($i=1,2,3,\dots$). Reversal error is represented by E . See FIGURE 2(a).

$$E = \bar{X}_{s1} - X$$

EXPERIMENT FOR THE LEAST INCREMENT TEST

This least increment test is applied on a linear axis with closed loop feedback. It is designed to operate less than 5m/min.

Inductance micrometer is fixed on the stationary along the movement direction of the linear axis, and make sure it's perpendicular to the end face of movement part and contact with this end face.

Running the linear axis make the movement part towards the inductance micrometer the ten positive increments, and then ten negative increments. Each step should stay for five minutes to wait for the linear axis stable to get accurate data. The displacement of the linear axis is recorded continuously in a computer, as shown in FIGURE 4.

The performance of machine tools is determined by many factors, including construction, control system, feed velocity, etc. The feed velocity and the position loop gain are the most common and important parameters in the operation and adjustment of machine tools. This experiment explores the influences exerted by the feed velocity V and position loop gain K on the least increment of the linear axis. Flow diagram of experiment is shown as FIGURE 5.

RESULTS AND ANALYSIS

Many experimental data on the linear axis are

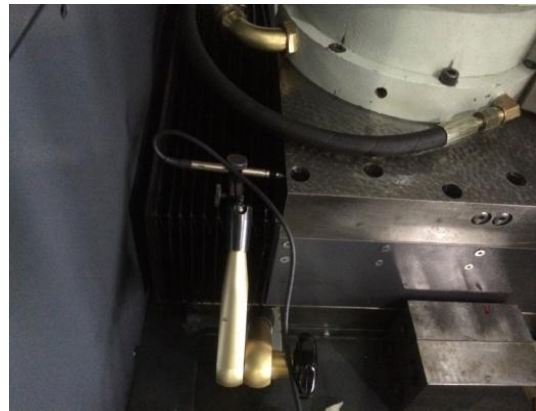
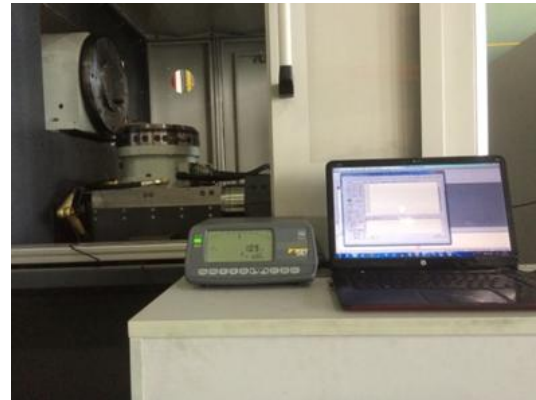


FIGURE 4. The linear axis and the setup for test

recorded including feed velocity from 1m/min to 5m/min and position loop gain from 0.7 to 1.3. Overshoot of each test is calculated according to the method described before.

Effect Of The Feed Velocity On The Least Increment Of The Linear Axis

FIGURE 6. shows the curve of velocity-overshoot.

the least increment of the linear axis is close to $3\mu\text{m}$ when the feed velocity is less than 2m/min, and the least increment of the linear axis is between $3\mu\text{m}$ to $4\mu\text{m}$ when the feed velocity is more than 2m/min. It is clearly that the feed velocity have no effect on the least increment under the allowed feed velocity of the linear axis.

Effect Of Position Loop Gain On The Least Increment Of The Linear Axis

FIGURE 7. shows that the least increment is $5\mu\text{m}$ when the position loop gain is set as 0.7, the least increment is between $2\mu\text{m}$ to $3\mu\text{m}$ when the position gain is set as 0.9 or 1, the least increment is close to $5\mu\text{m}$ when the position loop gain is set

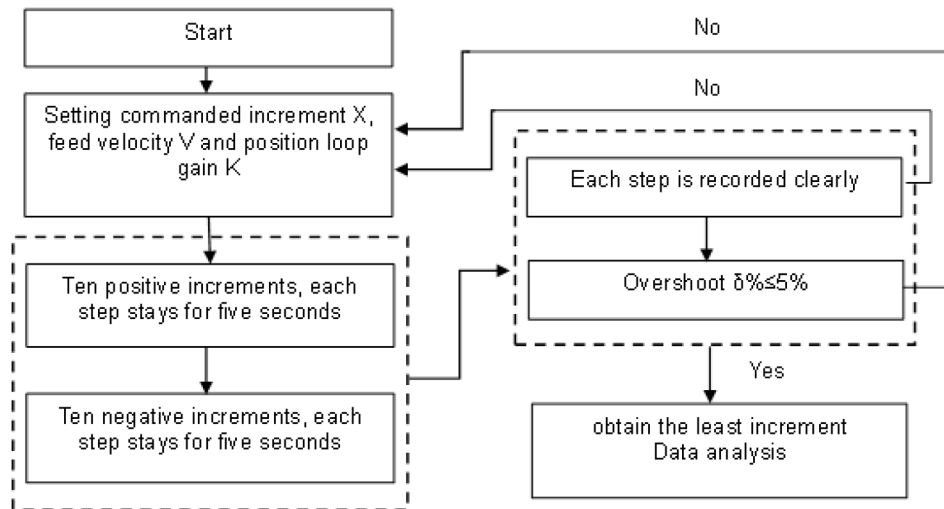


FIGURE 5. Flow diagram of the experiment

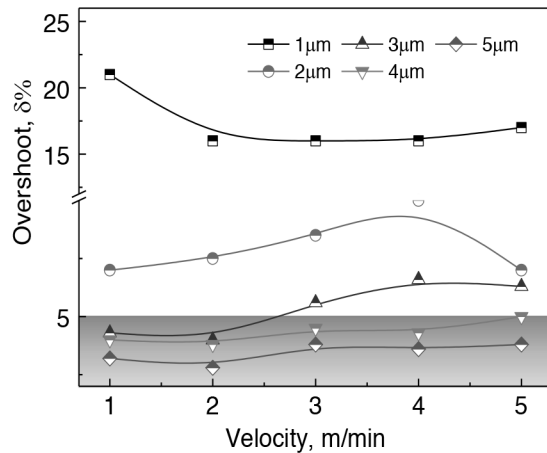


FIGURE 6. The curve of velocity-overshoot

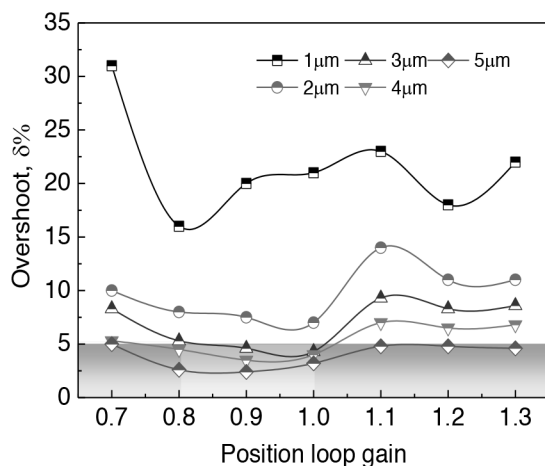


FIGURE 7. The curve of position loop gain-overshoot

as greater than 1. This indicates that the position loop gain have clear effect on the least increment of machine tools. it shouldn't be too large or too small, otherwise the optimum least increment of the linear axe couldn't be obtained. It is necessary to keep the position loop gain in a suitable range for the performance of machine tools.

The least increment test makes it easier to get the appropriate range that the position loop gain should be set. This test method is helpful for the precision adjustment of machine tools.

CONCLUSIONS

The least increment test that this paper proposed is a test method for the performance of machine tools. This test method could rapidly get the least increment and reversal error of machine tools.

The experiment conducted on a linear axe with closed loop feedback studied the effects of the feed velocity and the position loop gain on the least increment of machine tools. The results indicate that the feed velocity has no effect, but the position loop gain has an apparent effect on the least increment of machine tools.

The value of position loop gain at a certain range could make the machine tools obtain the optimum least increment. The least increment test makes it easier to get the appropriate range that the position loop gain should be set.

The least increment test could provide evidence for the precision adjustment of machine tools.

ACKNOWLEDGEMENTS

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HIGH PERFORMANCE DRY GRINDING OF CFRP WITH IN-PROCESS WHEEL CLEANING USING DRY ICE PARTICLE BLASTING

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INSTRUCTIONS

CFRP (Carbon Fiber Reinforced Plastics) is increasingly applied to airplanes, automobiles, production equipment for electronic device and so on. CFRP needs machining after molding process, and grinding is often used for edge finishing of CFRP parts. The machining of airplane CFRP parts is carried out under dry or coolant mist supply conditions though coolant has superior cooling and lubrication performance avoiding marked tool wear and delamination of CFRP.

The study deals with the high efficient dry grinding of CFRP. In dry grinding of CFRP, the wheel loading occurs in an early grinding process and leads the reduced surface quality of CFRP due to the delamination of carbon fibers. In the previous report [1], the authors have measured the grinding temperature at wheel contact area in dry grinding of CFRP, and it was much higher than the glass-transition temperature of CFRP. Once the loading occurs, the grinding ability might be recovered by dressing in which CFRP chips adhered on wheel surface are removed to project abrasive grains. However the dressing operation reduces the grinding efficiency.

We develop the dry grinding technique of CFRP with high quality and high efficiency using the dry ice particle blasting. The dry ice blasting is applied to the cleaning of dies, molds or some kinds of single-crystal wafer and has received a lot of attention as one of environmentally superior industrial technologies [2,3]. We have proposed the cleaning of loaded wheel surface in dry grinding of hard carbon using the dry ice blasting in previous report [4]. The recovery of grinding ability of loaded wheel is recognized by dry ice blasting as post-process. The dry ice blasting leads recovery of enough abrasive grain protrusion of vitrified bonded diamond grinding

wheel. Therefore the proposed technique have potential for high efficiency dry grinding of CFRP when the dry ice blasting removes CFRP chips adhered on wheel surface in grinding process. In this study, the effects of in-process dry ice cleaning on the wheel life extension and the quality improvement of ground surface are investigated experimentally, comparing the ground surface roughness, the grinding force, the grinding temperature, the wheel surface and the ground surface with the normal dry grinding of CFRP.

EXPERIMENTAL PROCEDURE

Figure 1 shows the experimental setup of dry grinding of CFRP with dry ice particle blasting. The dry ice particle blasting is applied to the wheel surface in grinding process of CFRP, in order to control of loading on wheel surface. The down-cut surface grinding of CFRP is repeatedly carried out with the setting depth of cut using the dry ice particle blasting. The dry ice particles are

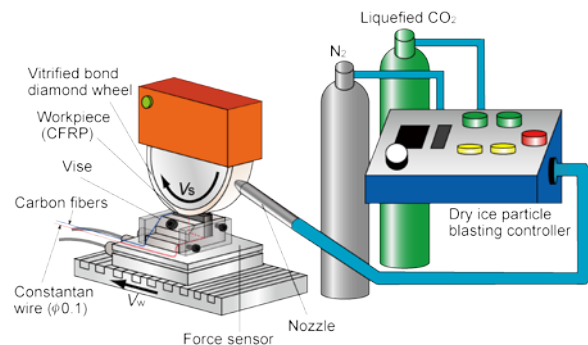


FIGURE 1. Experimental setup of dry grinding of CFRP with dry ice particle blasting to loading. Dry ice particles are generated from liquefied carbon dioxide and blast out with nitrogen assist gas.

TABLE 1. Main experimental conditions.

Grinding wheel	SD230F75V ($\phi 250 \times 10$ mm)
Workpiece	CFRP(UD) (Fiber angle $\theta = 0^\circ$, 20x20x5 mm)
Peripheral wheel speed V_s	23.6 m/s
Workpiece speed V_w	0.08 m/s
Setting depth of cut Δ	400 μm
Dry ice supply	16.7 g/s
Assist gas pressure	0.5 MPa
Nozzle standoff	1 mm

made of liquefied CO_2 in the blasting nozzle and are blown out with the high pressure assist gas of N_2 . The size of dry ice particle is less than 50 μm . The blast nozzle, whose exit diameter is $\phi 8$ mm, is fixed perpendicularly to the acting wheel surface with the standoff of 1mm at the setting position of coolant nozzle in normal surface grinding as shown in the figure. The unidirectional CFRP is selected as specimen and is ground parallel to the carbon fibers. The glass-transition temperature of the specimen is about 110°C by DSC[1].

The grinding forces and surface roughness are measured by the quartz force sensor (Kistler Type 9257B) and the surface roughness tester respectively. In addition the grinding temperature at the wheel contact area is measured by the developed thermocouple system consisting of carbon fibers in CFRP and constantan wires ($\phi 0.01\text{mm}$) [1]. The output (electro motive force) is about $0.41\mu\text{V}/^\circ\text{C}$. The ground surface of CFRP and acting wheel surface are observed by the optical microscope and SEM. Main experimental conditions are shown in table 1.

EXPERIMENTAL RESULTS

Grinding Force

Figure 3 show the variations of normal and tangential grinding force in repeated dry grinding processes under normal condition or in-process wheel cleaning using dry ice particle blasting. The grinding forces increase rapidly in early dry grinding process under the normal condition. The unusual sound and chattering occurs at about 320mm in grinding length under the normal dry grinding condition and the grinding length is defined as the limitation in normal dry grinding of CFRP. The grinding forces under the in-process dry wheel cleaning are approximately as same as those at 40mm in grinding length of

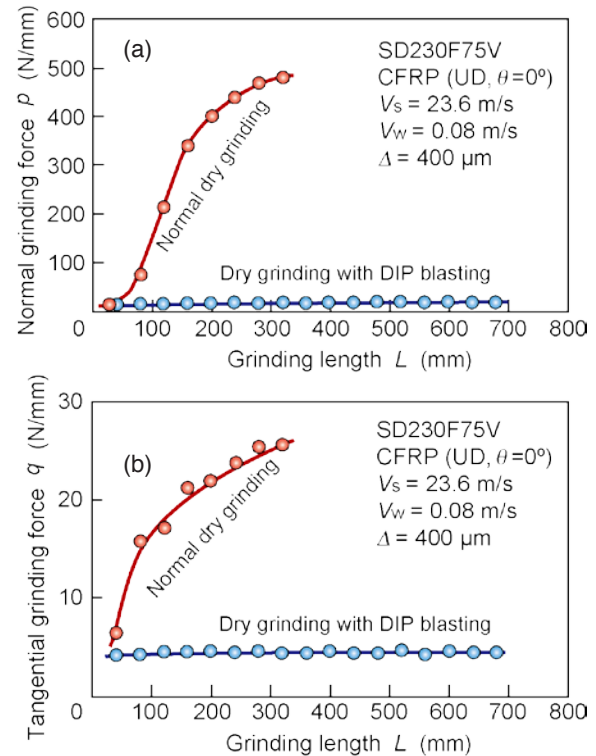


FIGURE 3. Variations of grinding forces (a: Normal force, b: Tangential force) in dry grinding with in-process cleaning and normal dry grinding of CFRP.

normal dry grinding. However the grinding forces in early grinding process continue for 680mm in grinding length under in-process wheel cleaning using dry ice particle blasting. Though the grinding tests with in-process dry ice blast wheel cleaning are carried out by 680mm in grinding length, the good grinding ability might continue in further dry grinding process. The tangential grinding force under in-process wheel cleaning is about 20% of that at grinding length limitation in normal dry grinding, and the normal grinding force is about 2% of that at grinding limitation in normal dry grinding.

Ground Surface and Wheel Surface

Figure 4 shows the optical photos and SEM images of ground surface of CFRP. The ground surface at 320mm in grinding direction under normal dry condition, when the grinding forces increase to become saturated, is shown in (b). The serious delamination of carbon fiber layers occurs, and lots of fractured carbon fiber chips are adhered on the ground surface. Therefore the normal dry grinding couldn't generate the ground surface of CFRP with enough surface quality in early grinding process. On the other

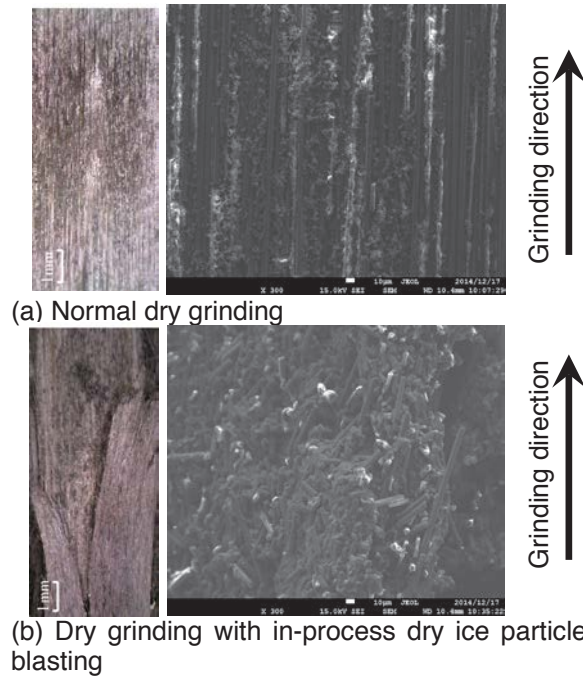


FIGURE 4. Ground surface of CFRP by dry grinding with in-process dry ice particle blasting and normal dry grinding.

hand, the ground surface with in-process wheel cleaning using dry ice particle grinding is shown in (a). There exists no delamination of carbon fibers on the ground surface, and carbon fibers appearing on the ground surface are arranged in an original direction of specimen. Therefore the dry grinding with in-process wheel cleaning achieves good quality in ground surface of CFRP.

Figure 5 shows the photographs of acting wheel surface in dry grinding processes. In the cases of normal dry grinding, the loading already occurs at 80mm in grinding length, and most

diamond abrasive grains are covered with adhered CFRP chips in early grinding process. Furthermore the acting wheel surface is covered with CFRP chips completely, and no diamond abrasive grain appears on wheel surface at 320mm in grinding length when the grinding forces increase to become saturated. On the other hand, in the case of dry grinding with in-process wheel cleaning using dry ice particle blasting, most diamond abrasive grains appear on the acting wheel surface at 80mm in grinding length just as the wheel surface after dressing. Furthermore the loading does not occur, and the acting wheel surface using the dry ice particle blasting is rarely different from the dressed wheel surface even at 680mm in grinding length, which is much larger grinding length than that at grinding limitation in normal dry grinding of CFRP.

Figure 6 shows the variation of ground surface roughness of CFRP R_a in both dry grinding processes. The surface roughness increases rapidly with an increase of grinding length and reaches about $10\mu m R_a$ at 120mm of grinding length in the normal dry grinding. Furthermore the delamination occurs over just 120mm in grinding length, and shaggy carbon fibers on ground surface prevent the measurement of surface roughness over 160mm in grinding length. In the meanwhile the ground surface roughness using dry ice particle blasting maintains about $2\mu m R_a$ in the dry grinding processes. From the viewpoint of acting wheel surface shown in figure 5, the loading might dominate the grinding results of CFRP, and the surface roughness about $2\mu m R_a$ would continue in additional dry grinding test of CFRP with in-process wheel cleaning using dry ice particle blasting.

	L=40mm	L=80mm	L=320mm	L=680mm
Dry grinding with in-process dry ice particle blasting				
Normal dry grinding				 200 μm same as on the left

FIGURE 5. Photographs of acting wheel surface in dry grinding with in-process cleaning and normal dry grinding of CFRP. No loading is observed on wheel surface with in-process cleaning using dry ice particle blasting.

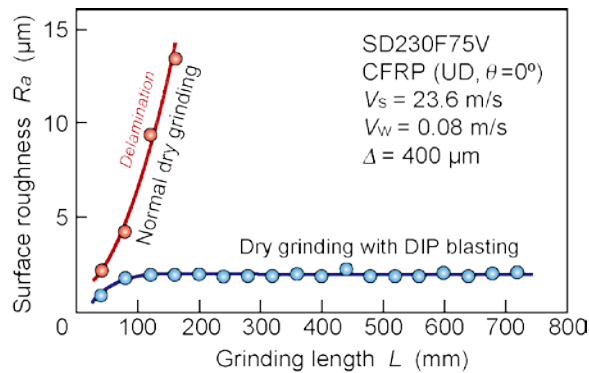


FIGURE 6. Variations of ground surface roughness in dry grinding with in-process cleaning and normal dry grinding of CFRP.

Grinding Temperature

Figure 7 shows the variations of grinding temperature at wheel contact area in both dry grinding processes. The grinding temperature at wheel contact area is determined by the maximum in variation of output from thermocouple fixed in CFRP specimens. The grinding temperature of about 250°C continues in dry grinding processes using dry ice particle blasting. However in the normal dry grinding processes, the grinding temperature is about 400°C at 40mm in grinding length, which is first grinding pass, and it increases over 700°C for a few grinding passes. The high grinding temperature leads the delamination on ground surface of CFRP in normal dry grinding. Though the grinding temperature is stable at about 250°C in the developed dry grinding using dry ice particle blasting, it is larger than the glass-transition temperature of the CFRP specimens. We'll investigate the thermal damaged layer beneath the ground surface of CFRP and develop the cool down technique during grinding processes when industrially necessary.

As outlined above, the in-process wheel cleaning using dry ice particle blasting makes the high performance in dry grinding of CFRP in both of grinding efficiency and quality.

CONCLUSIONS

The dry ice particle blasting affects the control of wheel loading and increase the dressing interval markedly. In particular, the wheel loading does not generate after the grinding at 700mm in grinding length using the dry ice particle blasting though the abrasive grains on wheel surface are covered with CFRP chips after 80mm in grinding length in normal dry grinding. In addition, the

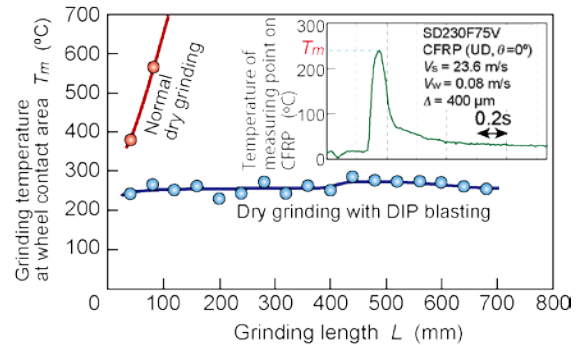


FIGURE 7. Variations of grinding temperature at wheel contact area in dry grinding with in-process cleaning and normal dry grinding of CFRP.

delamination of CFRP occurs in very early process of normal dry grinding. However the dry ice particle blasting assists smooth ground surface and extends the wheel life to more than 15 times of that in normal dry grinding. Thus the dry ice particle blasting to wheel surface could increase the wheel life in dry grinding of CFRP and effect the high quality and high efficiency grinding of CFRP.

FUTURE WORKS

The optimum conditions of dry ice particle blasting, which are nozzle standoff, blasting angle, and so on, for avoidance of loading are made clear, analyzing the blasting pressure on wheel surface. In addition its applications to grinding other difficult-to-machine materials are investigated.

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EXPERIMENTAL PLATFORM FOR IN-PROCESS METROLOGY DURING ORTHOGONAL TURNING

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INTRODUCTION

Chip formation mechanics in machining operations have been studied for several decades. These efforts aim to reduce tool wear, improve workpiece surface quality, and facilitate machining automation by reliably predicting and controlling chip segmentation length and frequency. It has been determined that the three main factors which impact chip formation mechanics are: 1) the chip thickness; 2) the radius of curvature of the chip; and 3) the mechanical properties of the workpiece material [1]. Because the chip thickness and radius of curvature are strongly influenced by the geometry of the cutting tool, carbide insert manufacturers provide a myriad of chip breaking geometries to suit specific machining applications. Additionally, researchers have implemented with the modulated tool path (MTP) approach to control chip segmentation length and frequency [2].

The aim of this study is to evaluate the chip formation characteristics for various cutting edge geometries and machining strategies in orthogonal cutting. Orthogonal cutting is approximated using a tube turning setup where a thin-walled, axisymmetric workpiece is machined along its axis. In-process metrology includes high-speed micro-videography, cutting force, tool-chip temperature, and tool wear measurements.

In this paper the experimental platform for studying chip formation in orthogonal turning, including the in-process metrology, is described. Representative high-speed video images, cutting forces, and cutting temperatures are presented to show typical results for an orthogonal turning experiment. Data are included for both traditional turning toolpaths, where the feed remains constant throughout the cut, and for modulated toolpaths (MTP), where the nominal tool motion is modulated using a low frequency/amplitude sine wave in the direction tangent to the machined surface. Finally, future

work is proposed where flexibility is added to the cutting tool to facilitate unstable machining operations (i.e., chatter). This novel approach will allow for the observation of chip formation processes during chatter at high magnification and frame rate.

EXPERIMENT SETUP

The test bed for the experiments was a Haas TL-1 CNC lathe (8.9 kW, 2000 rpm). Tubular workpieces were machined from AISI 1026 cold-drawn steel and 6061-T6 aluminum tubing. For brevity, only results from the AISI 1026 steel are reported here. The outside diameter of the workpieces was 72 mm and the wall thickness was 1 mm. Concentricity and cylindricity of the workpiece's outside and inside diameters with the rotational axis of the lathe spindle was assured by performing a finish turning operation immediately prior to conducting the experiments. Orthogonal tube turning was selected, rather than disc or flange turning [3-4], so that cutting speed, v_c , would not vary with a fixed spindle speed. Experiments were conducted with a cutting speed of 75 m/min at feedrates of {0.051, 0.102, 0.152, and 0.203} mm/rev.

The carbide inserts (Type C, 80° parallelogram) were provided by Kennametal. They are summarized in TABLE 1.

TABLE 1. Insert descriptions.

Insert image	ANSI catalog number (Kennametal part no.)	Description
	CCMW3252 (3757916)	7° relief angle, no chip breaker
	CCMT3252LF (3758169) CPMT3252LF (3755485)	7° and 11° relief angle, chip breaker
	CCMT3252MP (3744955) CPMT3252MP (3744958)	7° and 11° relief angle, chip breaker

Dynamic cutting forces were measured using a three-axis dynamometer (Kistler 9257B) rigidly mounted to the lathe's cross slide. A high-speed camera (Fastec IL-3) with a maximum frame rate of 1250 frames/sec and 5 mm field of view was mounted to a tripod which was fixed to the shop floor. An infrared camera (FLIR E40) was used to establish temperature trends with changes in machining conditions. A digital microscope was also attached to the lathe to measure flank wear between cutting tests. A photograph of the setup is provided in FIGURE 1 and additional details are provided in FIGURE 2.

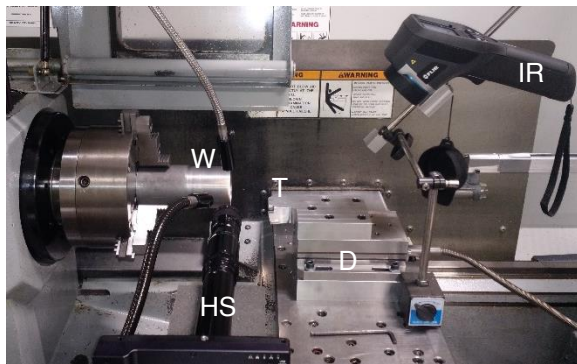


FIGURE 1. Orthogonal tube turning setup including: workpiece (W), tool (T), high-speed camera, (HS), infrared camera (IR), and dynamometer (D).

Cutting forces which occur during orthogonal turning can be described in a two dimensional coordinate frame. The force components are defined in the thrust, F_t , and cutting speed, F_c , directions as:

$$F_t = k_t b h \quad (1)$$

$$F_c = k_c b h \quad (2)$$

where b is the chip width (i.e., tube wall thickness) and h is the chip thickness (i.e., feed per revolution). The cutting force coefficients, k_t and k_c , are empirical constants which relate cutting forces to uncut chip geometry. The cutting force coefficients are primarily dependent on the workpiece material-cutting tool combination.

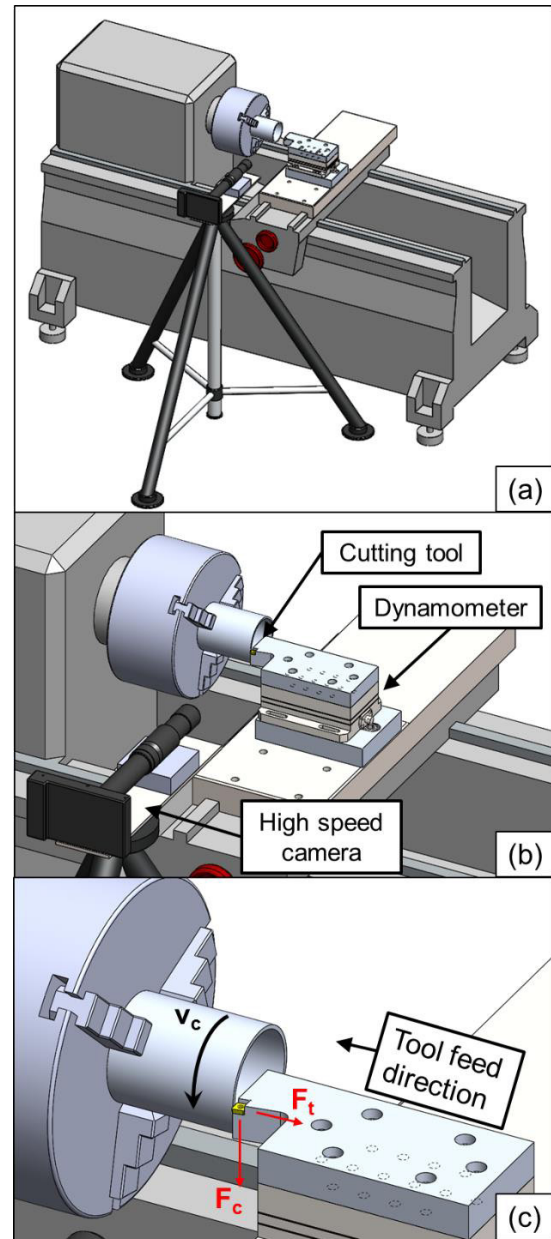


FIGURE 2. Tube turning setup: (a) shop floor perspective, (b) detail perspective, and (c) cutting force component directions (thrust, t , and cutting speed, c).

CONSTANT FEED EXPERIMENTS

As a representative example of the results from constant feed rate experiments, high-speed video images, dynamic cutting forces, and cutting temperatures are presented. The selected constant feed experiment was conducted at a cutting speed of 75 m/min with a feedrate of 0.203 mm/rev with a CCMW3252 type insert (i.e., no chip breaker). The workpiece

material was 1026 steel. FIGURE 3 shows the formation of a continuous chip for the orthogonal cutting operation.

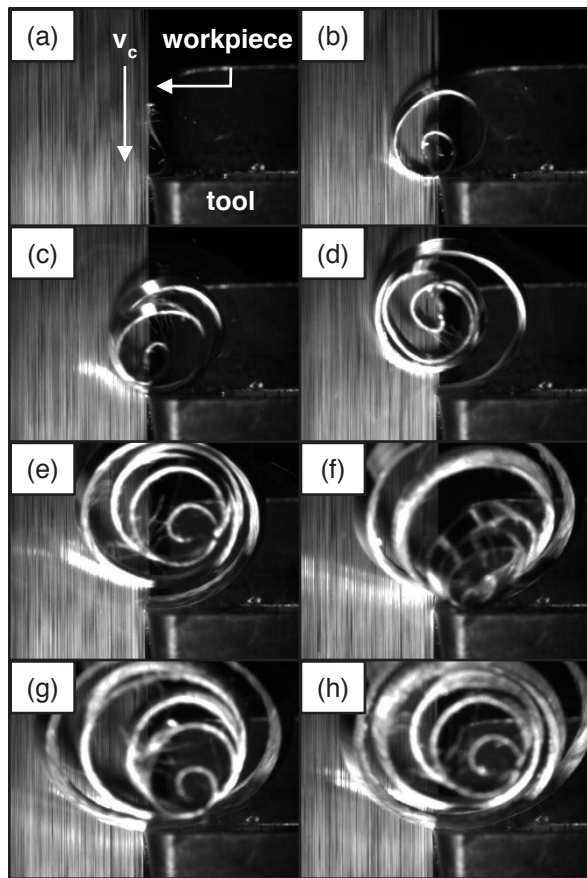


FIGURE 3. Progression of continuous chip formation in orthogonal cutting of 1026 steel.

The dynamic cutting forces for the operation are displayed in FIGURE 4. As the tool progresses through the cut, the force components increase rapidly until a quasi-steady state is reached. Because the chip geometry remains constant throughout the cut, the cutting force also remains constant. The thermal image, shown in FIGURE 5, indicates the maximum temperature for the constant feed operation.

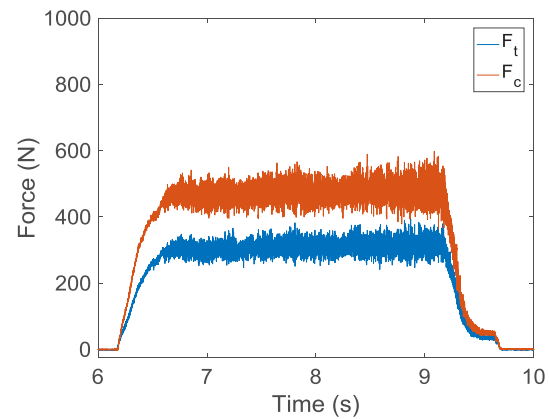


FIGURE 4. Cutting force components in the thrust and cutting speed directions for the constant feed turning operation.



FIGURE 5. Thermal image showing maximum cutting temperature for the constant feed turning operation.

For each insert geometry, the cutting force coefficients were calculated. The resulting thrust direction force coefficients, k_t , and cutting speed direction force coefficients, k_c , are shown in FIGURE 6 and FIGURE 7, respectively.

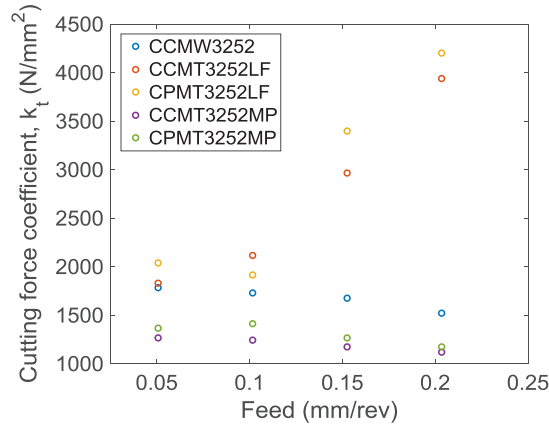


FIGURE 6. Thrust direction force coefficients for all insert geometries.

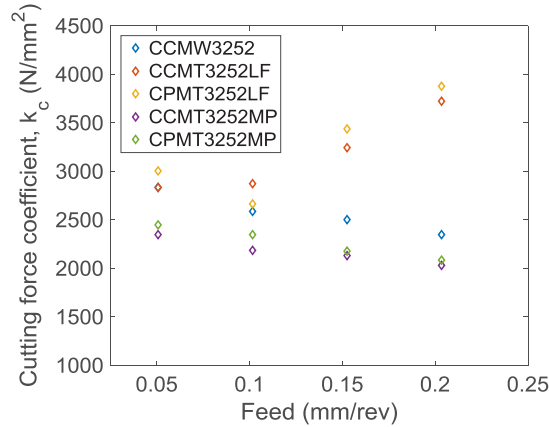


FIGURE 7. Cutting speed direction force coefficients for all insert geometries.

MTP EXPERIMENTS

Modulated tool path (MTP) is a turning technique which produces discontinuous chips by superimposing tool oscillations in the tool feed direction to repeatedly interrupt the cutting process [5]. An exaggerated example of an MTP turning operation is displayed in FIGURE 8 for illustrative purposes. The chip segmentation length is dependent on two, user-defined MTP parameters: 1) the tool oscillation frequency relative to the spindle speed, *OPR*; and 2) the oscillation amplitude relative to the global feed per revolution, *RAF*. The parameters are defined as:

$$RAF = \frac{A}{f_r} \quad (3)$$

$$OPR = 60 \cdot \frac{f}{\Omega} \quad (4)$$

where f_r is the global feed per revolution for a traditional, constant feed turning operation, A is the tool oscillation amplitude for an MTP turning operation, f is the tool oscillation frequency in Hz, and Ω is the spindle speed in rpm.

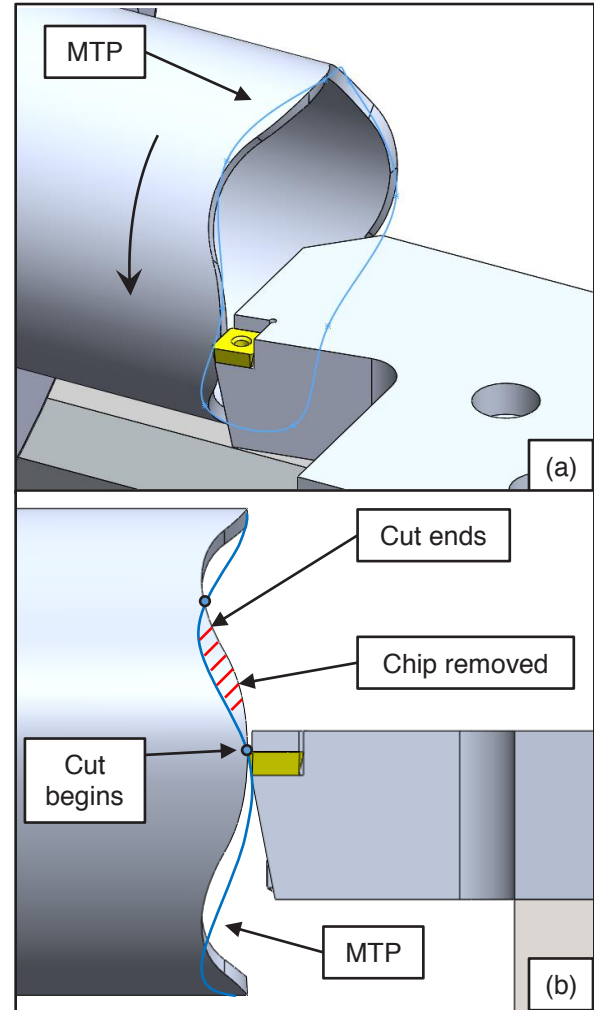


FIGURE 8. Illustrative example of: (a) an MTP turning operation; and (b) MTP chip formation.

As a representative example of the results for an MTP turning operation, high-speed video images, dynamic cutting forces, and cutting temperatures are presented. The machining parameters for the selected MTP experiment are summarized in TABLE 2. The workpiece material was 1026 steel.

TABLE 2. Machining parameters for the selected MTP turning operation.

Cutting speed, v_c	75 m/min
Global feedrate, f	0.203 mm/rev
RAF	0.8
OPR	0.5

FIGURE 9 shows the formation of a discontinuous chip during an MTP orthogonal turning operation. Adiabatic shear banding is observed in images (c)-(g). Because the tool oscillates in the feed direction (i.e., it enters and exits the cut), the instantaneous feedrate varies throughout the cut producing a chip of varying thickness. Similarly, the variation in chip thickness causes a subsequent variation in the dynamic cutting forces as shown in FIGURE 10.

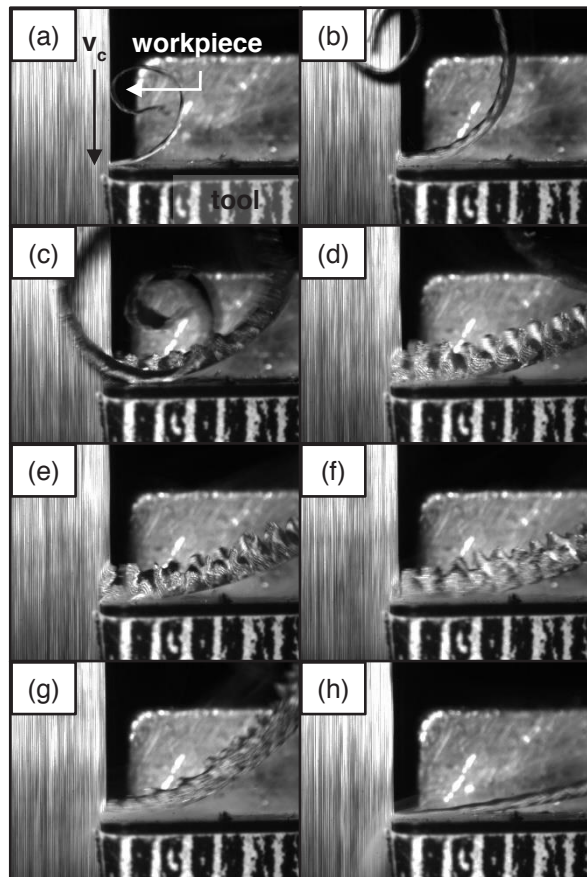


FIGURE 9. Progression of chip formation for an MTP orthogonal cutting operation.

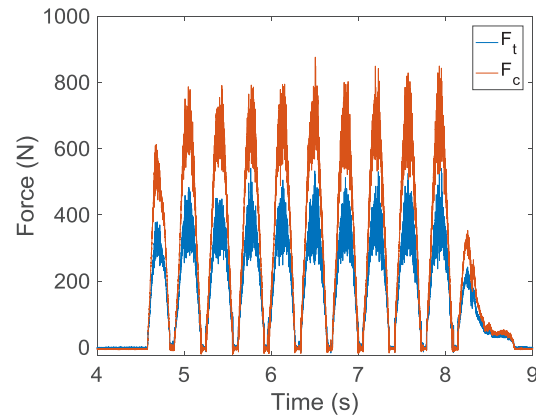


FIGURE 10. Cutting force components in the thrust and cutting speed direction for the MTP turning operation.

The thermal image shown in FIGURE 11 indicates the maximum cutting temperature for the MTP turning operation.



FIGURE 11. Thermal image showing maximum cutting temperature for the MTP feed turning operation.

TABLE 3. Maximum cutting force in the thrust and cutting speed directions.

	Constant feed	MTP
F_t (N)	330	396
F_c (N)	502	705

COMPARISON

To facilitate ease of comparison between the constant feed and MTP experiments, cutting forces, cutting temperatures, and chip

formations are presented in TABLE 3, FIGURE 12, and FIGURE 13, respectively.

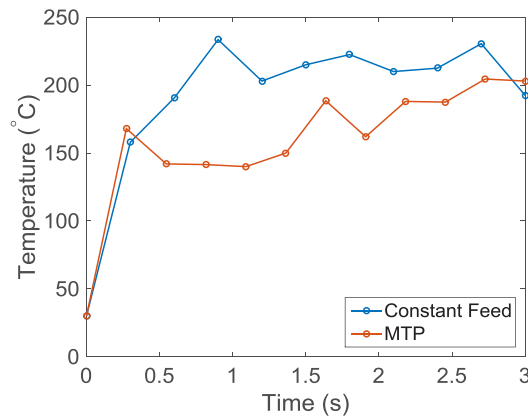


FIGURE 12. Cutting temperature increase during the constant feed and MTP turning operations.

SUMMARY AND FUTURE WORK

In this paper, an experimental platform for studying chip formation in orthogonal turning including in-process metrology was described. The metrology includes high-speed micro-videography, cutting force, tool-chip temperature, and tool wear measurements.

As an extension of the work presented here, high-speed videography of the chip formation process during unstable machining operations (i.e., chatter), will be investigated by adding flexibility to the tool in the feed and cutting speed directions. Further investigation of the adiabatic shear banding observed during the MTP experiments is also proposed.

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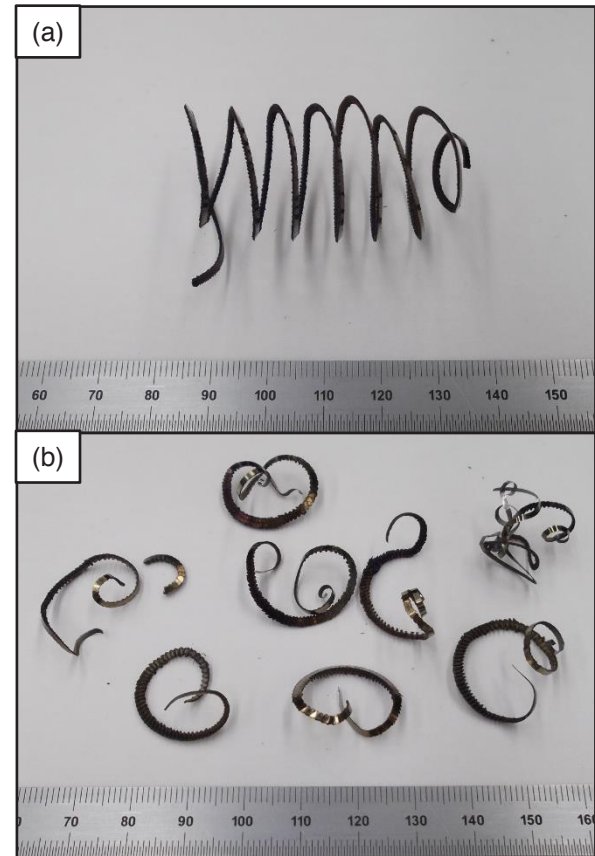


FIGURE 13. (a) Continuous chip for constant feed; and (b) discontinuous chips for MTP.

ABRASIVE POLISHING USING PLATE SPRING POLISHER FOR OPTICAL SURFACES

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INTRODUCTION

Some optics, such as the hollow-type corner cube mirrors used in the next lunar laser ranging system [1, 2], require smooth and highly accurate surfaces around their corners. However, it is difficult to finish such corners with high accuracy and smoothness by polishing with a conventional small tool, although such a tool can be effectively used to finish high-quality optical surfaces [3]. In general, small-tool polishing involves a rotational polishing tool. Such a tool is unsuitable for corners, making it difficult to polish them.

In our previous study, we proposed a new polishing method that uses a plate spring as a polishing tool [4]. In this method, a polishing pad is fixed to the tip of a plate spring, which moves back and forth on a workpiece surface. During the polishing, the plate spring is bent while maintaining a constant amount of deflection to generate a stable polishing load. In this manner, the edge of the plate spring enables the polishing of the corners of a surface.

The previous study showed that our method has one technical issue involving the bias of the polishing load applied on the workpiece surface. This bias causes nonuniform removal of the surface. To resolve this issue, in this study, we developed a new polishing tool that enables the application of a uniform polishing load on a polishing area. In this paper, we introduce the mechanism of the new polishing tool, and we demonstrate its fundamental characteristics in polishing experiments on silicon plates.

PRINCIPLE OF POLISHING METHOD

Figure 1 shows a schematic of the polishing method proposed in the previous study, illustrating, as an example, the case of polishing a corner cube mirror. As shown in Fig. 1, our method uses a plate spring that is bent to generate a polishing load. The polishing load is controlled by changing the amount of deflection of the plate spring. A polishing pad is fixed at the tip of the plate spring and moves on the

workpiece surface to polish it. Since the polishing pad is fixed on a rigid plate, our method is expected to easily polish areas around corners.

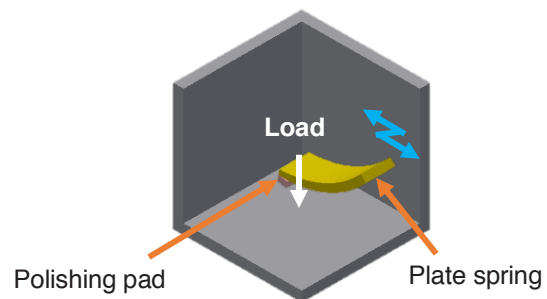


FIGURE 1. Polishing method using plate spring. The case of polishing a corner cube mirror is shown.

POLISHING MACHINE

Figure 2 shows the polishing machine we developed on the basis of the principle discussed in the previous section. In this machine, a polishing pad is fixed at the tip of a plate spring and is placed in contact with the workpiece surface via a vertical stage. A load cell placed under a worktable detects the polishing load. A numerically controlled stage is placed under the load cell to move the workpiece.

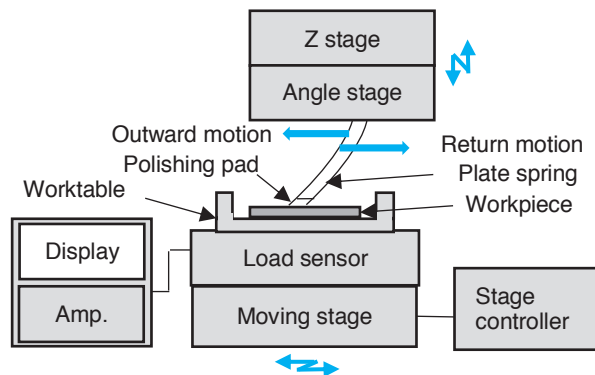


FIGURE 2. Schematic drawing of polishing machine using a plate spring.

To polish the surface using the machine, the plate spring is first moved downward by the vertical stage so that it is bent, which generates a polishing load on the surface. Then, while maintaining this situation, i.e., the bending of the plate spring and the polishing pad in contact with the surface, the workpiece is moved by the workpiece-driving stage under numerical control, allowing the surface to be polished.

POLISHING EXPERIMENTS USING A RIGID-TYPE POLISHING TOOL

In this paper, we refer to the previous polishing tool as a rigid-type polishing tool. To assist understanding in the present paper, in this section, we briefly describe the previous experimental results using the rigid-type polishing tool. We also discuss a technical issue of the polishing tool.

Experimental method

Figure 3 shows the rigid-type polishing tool. Using the polishing tool, we polished single-crystal silicon plates as workpieces and examined the shapes of the polished areas. In the polishing experiment, the polishing tool was moved back and forth on a workpiece surface while moving the workpiece with a stage. To accurately evaluate the removal profile by excluding the effect of the flatness error of the initial surface, the number of round trips of the polishing tool was determined such that the removal depth was over approximately $10\text{ }\mu\text{m}$. The silicon plate used in the polishing experiments was a silicon wafer with a thickness of 0.7 mm roughened by sandblast.

The resulting surface profiles were measured using a laser probe three-dimensional measuring instrument [5]. The polishing conditions were as follows: a polishing load of 26 N , a polishing speed of 130 mm/s , and a distance of polishing tool movement of 17 mm . The polishing grains were silicon carbide with an average grain size of $3\text{ }\mu\text{m}$ with a concentration of 10% in tap water.

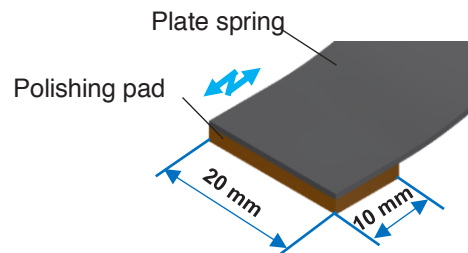


FIGURE 3. Tip of rigid-type polishing tool.

Experimental result

Figure 4 shows the profile of the polished surface in the direction perpendicular to the polishing tool motion with the initial surface profile. It can be seen that the removal depth is nonuniform. To obtain a flat surface by our polishing method, the removal depth resulting from a single round trip should be uniform, unlike the profile shown in Fig. 4. One possible reason for such nonuniform removal is that a nonuniform polishing load was applied on the polished area during the polishing process. Thus, we investigated the polishing load applied on the surface, as described in the next section.

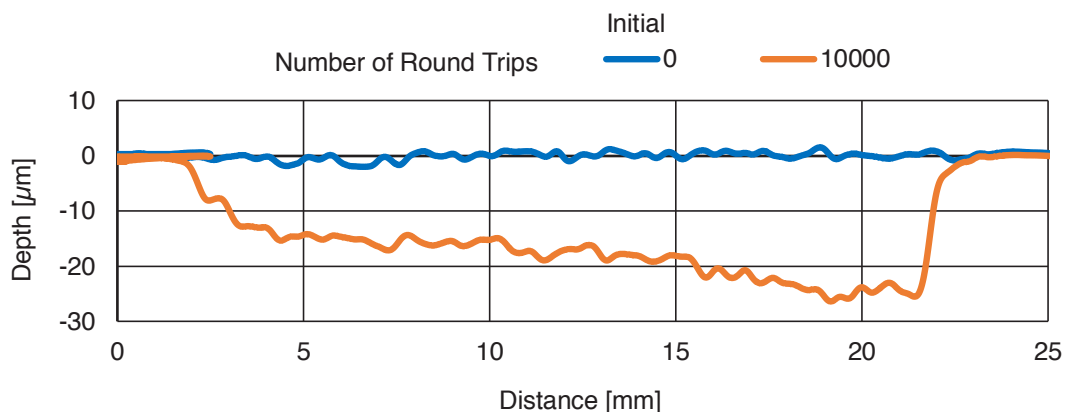


FIGURE 4. Profiles of surface before and after polishing in the direction perpendicular to tool motion. Polishing was conducted using the previous polishing tool, i.e., the rigid-type polishing tool.

MEASUREMENT OF POLISHING LOAD DISTRIBUTION

Experimental method

If there is bias in the polishing load, the removal depth may vary as shown in Fig. 4. To confirm the existence of bias in the polishing load, we investigated the distribution of the polishing load by using a pressure measurement film, whose color changes from white to red when pressure is applied on its surface, where the color density of red varies with the amount of pressure. In the investigation, the pressure measurement film was placed on the worktable of the polishing machine and pressed by the polishing tool. After pressing for 2 min, we observed the density distribution of the color.

Experimental result

Figure 5 shows a photograph of the film after pressing by the polishing tool. As shown in the figure, the color density is different at both ends of the colored area, and the width of the colored area is also different at both ends. This observation shows that the polishing load is nonuniform. Furthermore, from Figs. 4 and 5, it is found that the removal depth increases with increasing color density and width of the colored area. This finding proves that there is bias in the polishing load for the rigid-type polishing tool. Therefore, to improve the uniformity of the removal depth, we developed a new polishing tool that applies a uniform polishing load.

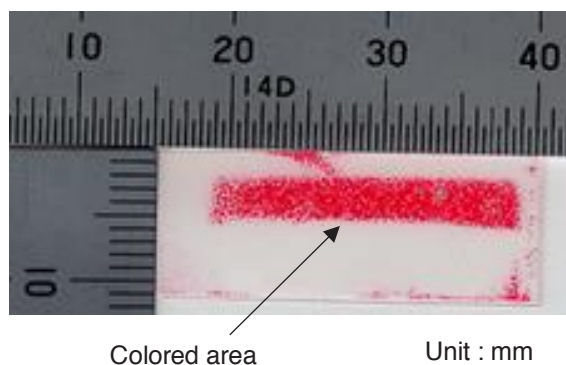


FIGURE 5. Observation of pressure measurement film after pressing by the rigid-type polishing tool.

EXPERIMENTS ON POLISHING TOOL APPLYING UNIFORM POLISHING LOAD

Mechanism of new polishing tool

To achieve uniform removal, we developed a new polishing tool that applies a uniform polishing load,

as shown in Fig. 6. In the new polishing tool, the polishing pad is not directly fixed on the plate spring, unlike in the rigid-type polishing tool, but on the bottom surface of a U-shaped block. A polishing load is applied to the polishing pad through the U-shaped block by a plate spring. A pin is installed on the U-shaped block as shown in Fig. 6. The plate spring presses down the top of the pin to apply a load to the polishing pad via the U-shaped block.

This arrangement allows the point contact of the plate spring with the U-shaped block, which results in the application of a uniform polishing load on the workpiece surface, even though the plate spring is inclined. Moreover, the polishing pad can move while being in close contact with the workpiece surface because the plate spring is connected with the U-shaped block, which has a high degree of freedom, although the rotation of the U-shaped block around the Z-axis is prevented. Thus, even when the polishing tool moves near the vertical surface of the corner cube mirror shown in Fig. 1, the U-shaped block does not collide with the vertical surface.

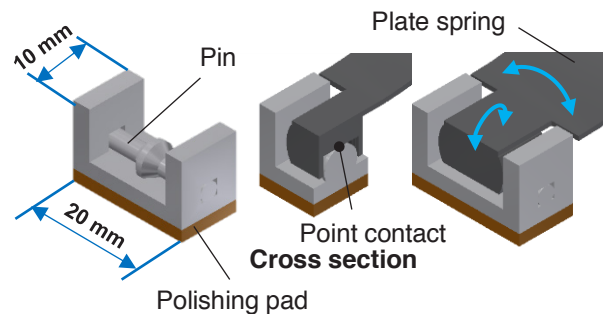


FIGURE 6. Newly developed polishing tool.

Polishing experiment

To demonstrate the effectiveness of the new polishing tool, we polished single-crystal silicon plates and measured the removal depth of the polished surface. The polishing and measurement methods were the same as those used in the experiment on the rigid-type polishing tool.

Figure 7 shows the surface profile of a polished area in the direction perpendicular to the tool motion with the initial surface profile. As shown in the figure, the removal depth is uniform. This result indicates that the new tool applies a uniform polishing load.

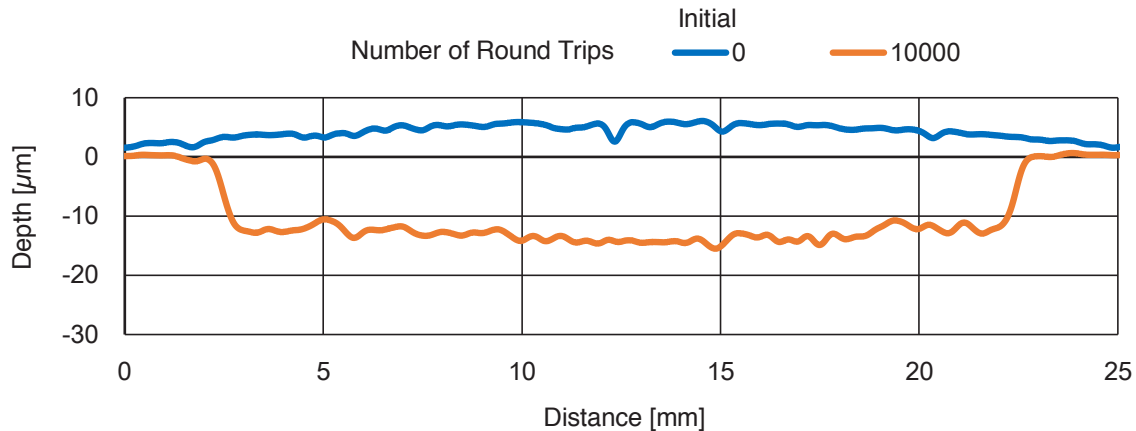


FIGURE 7. Profiles of surface before and after polishing in the direction perpendicular to tool motion. Polishing was conducted using the newly developed polishing tool.

CONCLUSIONS

In this study, we improved the removal characteristics of our previously proposed polishing method; the removal depth obtained by the previous method was nonuniform.

In the present study, the polishing load distribution was investigated, which revealed that the previous rigid-type polishing tool had a nonuniform polishing load, leading to the nonuniform removal of polished areas. On the basis of this finding, a new polishing tool using a plate spring was developed to apply a uniform polishing load. By using the new tool, we successfully polished silicon plates with a uniform removal depth.

ACKNOWLEDGEMENTS

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Design and experimental analysis of step drills aiming orthopedic surgery

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INTRODUCTION

Drilling is a common requirement in orthopedics, fracture treatment and reconstructive surgery. Although widely used, drilling is still the cause of several surgical complications, such as bone breakthrough and bone necrosis, mainly caused by high thrust forces (F_{th}), torques (M_z), and temperature (T) during bone drilling [1]. In order to try to solve those problems, experimental and numerical optimizations from surgical standpoint have been performed by changing the drill point angle, flutes, drill diameter, drill speed, and feed rate, related to F_{th} , M_z , and T [2]. Although the performance has been improved, there are issues intrinsic to the structure of standard drills, and a comprehensive analysis of these parameters is still necessary before surgical requirements can be achieved [3].

In this article, based on the bone structural and thermal characteristics, a step drill was designed by the mechanics and thermodynamic analysis of the drilling process. Experiments were carried out on a drilling system by altering the structural parameters to achieve optimized performance.

STEP DRILL DESIGN

Lee, *et al.* [4, 5] and Sui, *et al.* [6, 7] presented mechanistic and thermodynamic models for the prediction of F_{th} , M_z and, T during bone drilling, which was mainly generated from three sources: drill lips, chisel edge and margins. So F_{th} , M_z and T can be expressed as:

$$F_{th} = F_{thc} + F_{thlip} + F_{thmar} \quad (1)$$

$$M_z = M_c + M_{lip} + M_{mar} \quad (2)$$

$$T = T_c + T_{lip} + T_{mar} \quad (3)$$

where, F_{thc} , M_c and T_c arise at the drill lips, F_{thlip} , M_{lip} and T_{lip} arise at the drill chisel, F_{thmar} , M_{mar} and T_{mar} arise at margins. However, F_{thmar} , M_{mar} and T_{mar} arising at margins is very small which can be neglected [8, 5]. From the oblique cutting assumption, F_{th} , M_z and Heat (Q) of the drilling can be calculated by:

$$F_{th} = 2 \int_{r_0}^R dF_z + 2 \int_{r_{ind}}^{r_0} dF_{z_sec} \quad (4)$$

$$M_z = 2 \int_{r_0}^R dM_z + 2 \int_{r_{ind}}^{r_0} dM_{z_sec} \quad (5)$$

$$Q_T = 2 \int_{r_0}^R dQ_{shb}(r) + 2 \int_{r_{ind}}^{r_0} dQ_{fd}(r) \quad (6)$$

Where R is the drill outside radius, r_0 is the minimum value of r along the cutting lips, r_{ind} is the radius of the indentation zone, F_{z_sec} and M_{z_sec} are the force and torque arising at the secondary cutting edge, $Q_{shb}(r)$ is the amount of heat transferred into bone, $Q_{fd}(r)$ is the amount of heat transferred into drill. It can be found that F_{th} , M_z and Q_T from the two sources are positively correlated with diameter. Based on those results, the pre-drilling procedure enlarging the drilled diameter gradually by a series of drills can be put forward. It should more equally distribute the generation of heat during the procedure, which reduces the amplitude of the high peak T seen in drill with standard shape.

Osseous tissue can be divided into two groups, cancellous bone and cortical bone, with the latter being the hardest kind. The time of F_{th} , M_z and T affecting on the bone mainly depends on the thickness of cortical bone, which is around 6.5~8mm for human's femur [9]. Thus, the step drill, which has been widely used in drilling the counter bore for metal, can be used instead of the processing by two drills to reduce time and complexity of the surgery, shown in Figure 1. Moreover, the thermal conductivity of cortical bone (0.38 W/mK) is much smaller than that of drills (16.2 W/mK). So a larger percentage of heat is mainly transferred by the drill, and the heat affected zone ($>47^\circ\text{C}$) on bone is relatively small, which generated by the first-step drill can be drilled by the second-step drill. The eventual heat affected zone is actually smaller due to tiny volume of bone removed by the second-step drill.

With the prior discussion, we conclude that the length of the first-step drill (L), and the difference between the diameters of the first-step drill (D_1) and the second-step drill (D_2) are the crucial factors in the design, which are evaluated in the

experiments. A visualization of L , D_1 , and (D_2) can be referred to Figure 1.

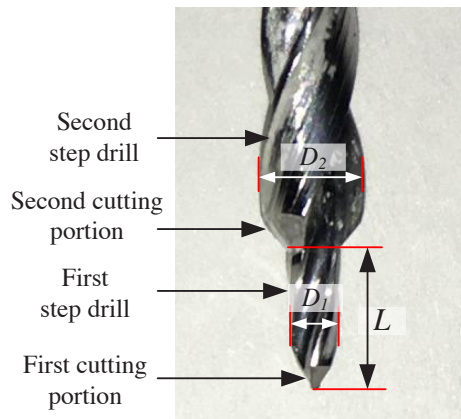


FIGURE 1. The newly designed step drill

EXPERIMENTAL METHOD

Preparation of specimens

The specimens of cortical bone were prepared along the longitudinal direction on a fresh bovine bone (age 3-4years) as it has similar mechanical and thermal properties of human bone. The process of sample production was shown in figure 2. The specimens were cut into 30mm (length) $\times$ 20mm (width) $\times$ 8mm (thickness) cuboids that were cooled by a refrigerated saline solution. Besides, in order to understand the influence of cancellous bone, the sawbone 1522-505 with a similar material property with human's femur was used in experiments. The sawbone specimens were cut into 30mm (length) $\times$ 20mm (width) $\times$ 15mm (thickness).

Furthermore, the black-body treatment was applied on samples for increasing the measurement accuracy of temperature by thermography.



FIGURE 2. Sample production process

Experimental setup

The drilling experiments were carried out on an OKKVM4-2 Machining Center, and an experimental setup measuring the thrust force, torque and temperature was designed and assembled. The device can be shown in Fig.3. A Kistler Dynamometer 9272 with charge amplifier 5070 was used to measure F_{th} and M_z . The data were sampled at 1–3 kHz. T was recorded by an infrared thermography (NEC Avio R300). The emissivity and sampling frequency were set as 0.94 and 50Hz, respectively.

Design of experiments

As it's impossible to measure T on the interface between drill margin and hole surface, an offset (0.5mm) was set from the interface. The basic geometry parameters of step drills were shown in Table 1. Eight prototypes of drills with different L , D_1 and θ were manufactured by TOKO Co., LTD, as shown in Table 2.

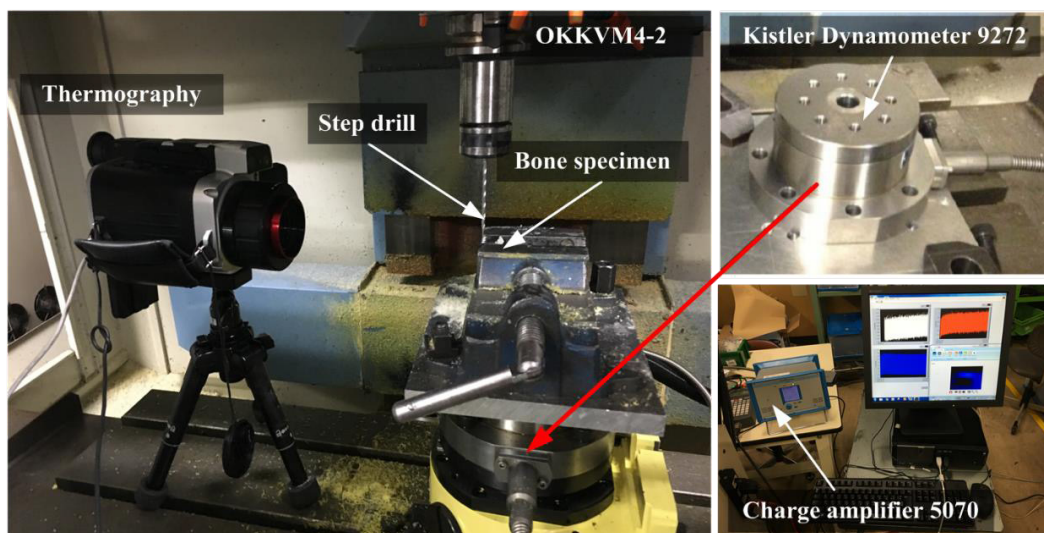


FIGURE 3. The experimental setup

TABLE 1. Basic geometry parameters of drill

Material	SUS402J2
Helix angle	28
Web thickness	17%
Chisel edge angle	95°
Point angle	90°

TABLE 2. Geometry parameters of drilling experiment

D_2 (mm)	D_1 (mm)	L (mm)
3.2	0	0
	1.2	5
	1.7	
	2.2	
	2.7	
	1.7	10

RESULTS and DISCUSSION

Effect of the diameter difference

The effects of D_1 on F_{th} , M_z and T are presented in Figure 4. It can be found that there are similar tendencies that F_{th} , M_z and T decrease followed by increasing with the increase of D_1 . An optimal result can be observed. When D_1 is 2.2mm under the current experimental condition, where F_{th} , M_z and T respectively reduce 20.2%, 8.8% and 25.0%, compared with standard drills.

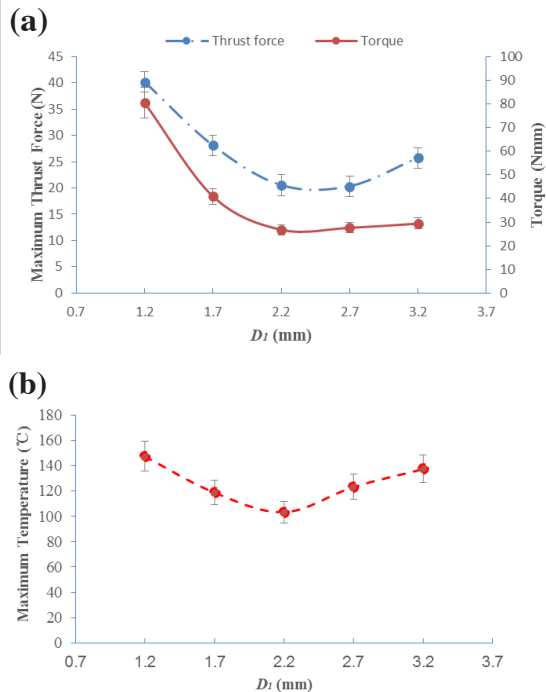


FIGURE 4. (a) Effect of D_1 on F_{th} and M_z ; (b) Effect of D_1 on T ;

Effect of the length of the first-step drill

For the same D_1 (1.7mm), the influences of L on F_{th} , M_z and T were studied. The results, seen Fig.5(a), show that F_{th} and M_z increase followed by the decrease with L . However, T decreases with L . The significant change that F_{th} , M_z and T respectively reduce 51.4%, 48.3% and 36.8% compared with standard drills can be found when L is larger than the bone's thickness.

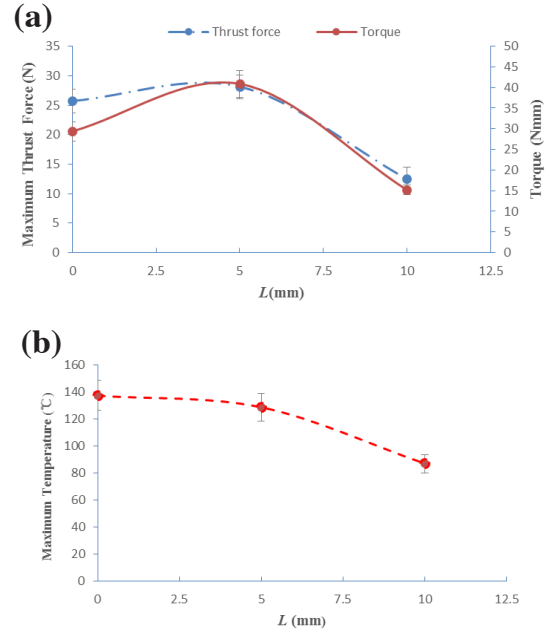


FIGURE 5. (a) Effect of L on F_{th} and M_z ; (b) Effect of L on T .

In order to understand the remarkable difference among the three drills, the variation of F_{th} during drilling was analyzed in Fig.6. For standard drills, as shown in Fig.6(a), the change of F_{th} can be mainly divided into four zones. In zone a, after the chisel edge contacts to cortical bone, cutting lips are gradually engaged in cutting and F_{th} gradually increases correspondingly. After both the chisel edge and cutting lips drill into cortical bone, F_{th} comes to be permanent (zone b). Due to the limited thickness of cortical bone (8mm), F_{th} decreases with the tip drilling into cancellous bone (zone c). When the drill gets through cortical bone, F_{th} is mainly generated by cancellous bone (zone d). It can be found that F_{th} during drilling cancellous bone (zone d) is much smaller than that of cortical bone (zone b). As to the step drill with 5mm first-step, F_{th} can be divided into eight zones, as shown in Fig.6 (b). F_{th} shown in zone b_1 is caused by the chisel

edge and cutting lips of first-step drill. F_{th} shown in zone b_3 is caused by cutting lips of second-step drill. The maximum F_{th} is achieved in zone b_2 , which is the addition of F_{th} in zone b_1 and zone b_3 . The slight vibration appeared in zone b_2 may be explained by many reasons, mainly because the bone is not completely immobilized and worktable vibrates with forces during drilling. In Fig.6(c), there is no addition appearing in zone b due to the limited thickness of cortical bone. The shape increase can be found when drill tip gets through cortical bone, which may be explained by the bending of specimens.

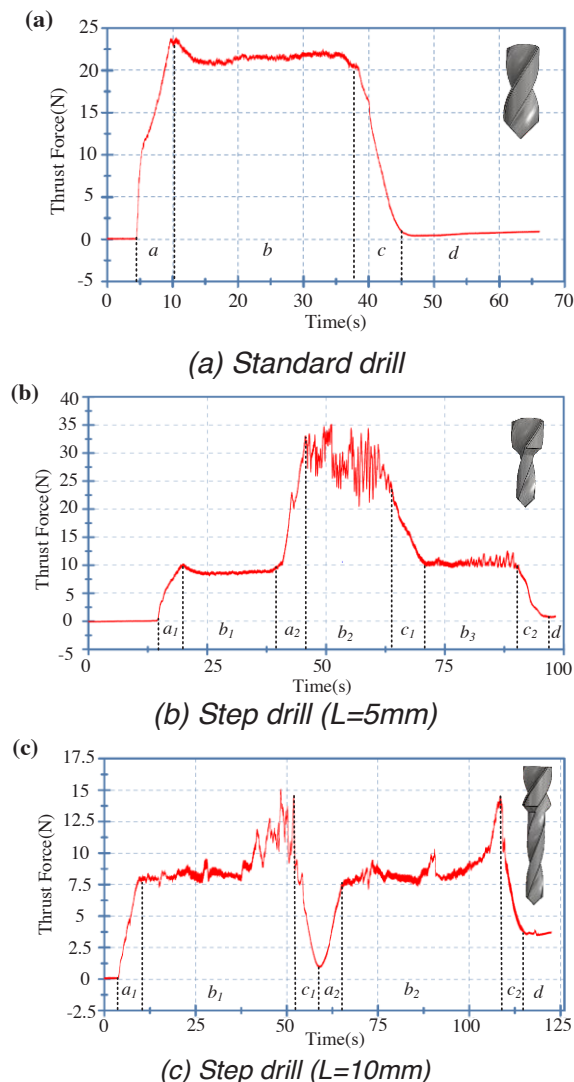


FIGURE 6. Evaluation of thrust force on three type of from drill bit by experiments

Fig.7 shows the temperature distribution on the offset interface of specimens by thermograph in bone drilling. For standard drills, the maximum T (137.4°C) arise at cutting lips, as shown in

Fig.7(a)(I). What's more, the temperature drops significantly when drill tip gets through cancellous bone, which is still larger than threshold of thermal damage. For the step drills, the maximum T arise at second-step cutting lips. Comparing with Fig.7(b)(I) and Fig.7(c)(I), we can found that the maximum T of step drills with 10 mm L is higher than that of the drill with 5 mm L , which is mainly caused by the heat accumulation due to the low thermal conductivity of cortical bone. However, the maximum T during the whole drilling period of step drills with 10 mm L is smaller than that of the drill with 5 mm L , as shown in Fig.7(b)(II) and Fig.7(c)(II).

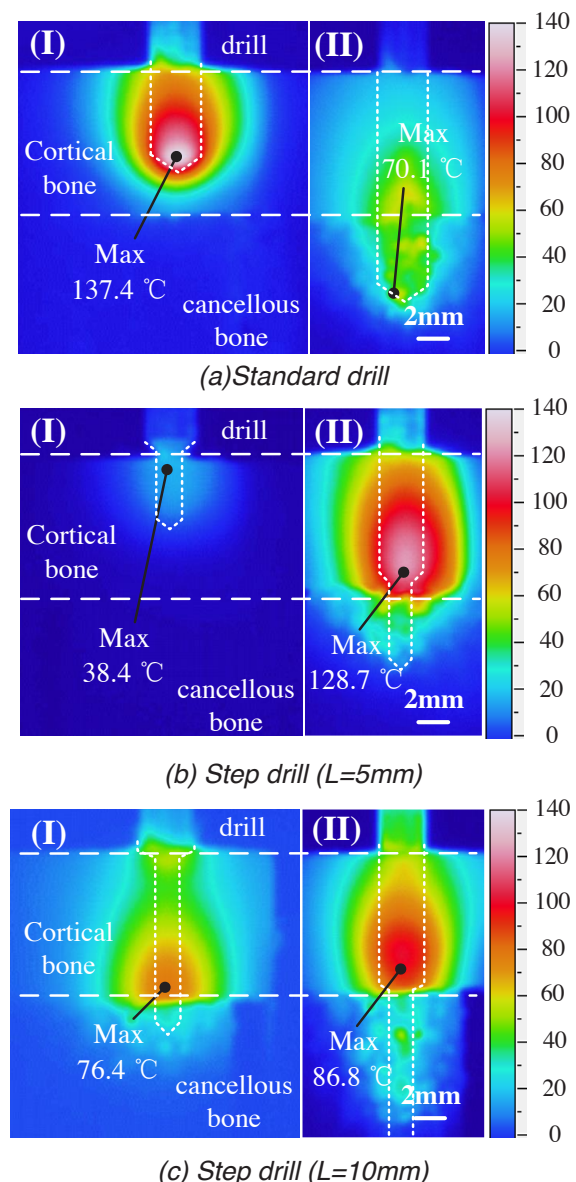


FIGURE 7. Thermographs in bone drilling

CONCLUSION

A novel design of step drill applied in orthopedic surgery has been evaluated in experiments of bone drilling. The relationships between F_{th} , M_z and T and the drills' structural parameters (D_1 and L) have been also obtained and analyzed. It can be found that the proposed design can not only reduce the mechanical and thermal damage in bone drilling in comparison with standard drills, but also be closer to meet the low-trauma requirements of orthopedic surgery.

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Study on construction of laser assisted crackless machining for Polybenzimidazole

(The influence of different laser irradiation conditions on machined surfaces by using orthogonal cutting)

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INTRODUCTION

Aromatic polybenzimidazoles (PBIs) have exceptional thermal and oxidative stability despite being thermoplastic resins. PBIs were developed for aerospace and defense applications both as nonflammable and thermally stable textile fibers and as high-temperature matrix resins, adhesives, and foams with the cooperation of the National Aeronautics and Space Administration (NASA) and the Air Force Materials Laboratory (AFML) [1]. Although inorganic fibers such as glass and boron nitride fibers also have a high heat resistance, organic fibers with greater extensibility and fusibility are superior to inorganic fibers for applications requiring such characteristics, such as in heatproof suits [2]. Among resins, polymeric materials composed of aromatic and heteroaromatic ring structures as chain constituents, like PBIs, have valuable engineering properties, such as flame retardance, excellent mechanical strength, and high strength retention over a wide range of temperatures, as well as toughness, chemical inertness, and adhesion. PBIs are thus suitable for many applications, including as matrix resins in structurally reinforced composites for aircraft and aerospace vehicles, as heat- and sound-absorbing materials for electronics designs, and as structural adhesives [3]. In addition, because the linear expansion coefficient of PBI-sintered materials is equal to those of aluminum alloy and brass ($23 \times 10^{-6} \text{ K}^{-1}$), it can be used together with metals [4].

PBI-sintered materials are produced using extremely high-pressure and high-temperature methods, such as hot-press processing, because of their high melting point [5]; thus, it is difficult to form complex shapes and minute components. In general, a second round of

processing that involves cutting is required to form the shape of the product. However, traditional cutting methods often produce many large cracks on the machined surface, causing the strength of PBI products to decrease [6]. To reduce the formation of cracks, a previous study focused on the softening temperature (650 K) and glass transition temperature (700 K) of PBI, and the cutting temperature was set to between these two temperatures [6]. Even though this approach was successful, it required extremely high cutting speeds (40 m/s) to ensure the cutting temperature was greater than 650 K; therefore, it is difficult to use this method as a general machining technique.

In this study, a laser-assisted machining (LAM) method that enables the crack-free processing of PBI at slower cutting speeds was developed using an orthogonal tool.

EXPERIMENTAL APPROACH

PBI was machined using a 25-W semiconductor laser unit and a single-crystal diamond orthogonal tool installed on a machining center. A schematic of the LAM setup is shown in Figure 1. The beam has a near-infrared wavelength of 917 nm and a top-hat distribution. The laser is integrated with a precision three-axis stage using an optical fiber. The distance between the end face of the fiber and the workpiece can be adjusted to vary the spot size of the laser. The only stationary component is the workpiece, which is mounted on the stage. All other components, including the tool bit and laser, move along the y-axis of the machining center. However, the tool bit and laser do not move relative to one another during machining. The laser controller is used to modulate the laser power. The workpiece has projections of 2 mm in width. The laser beam irradiates the

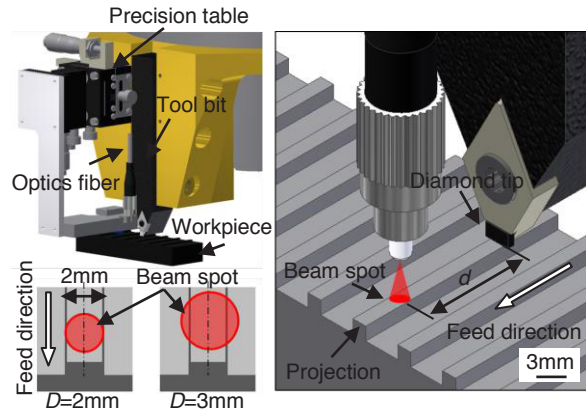


FIGURE 1. Schematic and actual view of laser-assisted machine tool.

TABLE 1. Experimental conditions.

Feed rate: F [mm/min]	500	
Depth of cut: A_d [mm]	0.05	
Cutting edge width of the cutting tool [mm]	3	
Rake angle [°]	0	
Cutting environment	dry	
Laser spot diameter: D [mm]	2	3
Distance between irradiated area and cutting point: d [mm]	6, 15	6
Laser output: Q [W]	0~19.3	
Dimensions of the workpiece [mm]	90 × 30 × 15	

surface of the projection. The conditions for each experiment conducted in this study are given in Table 1. The machined surfaces of the workpiece after the experiments were observed using digital microscopy and scanning electron microscopy (SEM) to confirm the presence or absence of cracks. Fourier transform infrared (FTIR) spectroscopy was used to investigate the thermally affected layer of the machined surface.

FINISHED SURFACE AND THERMAL EFFECT AFTER LASER-ASSISTED MACHINING

Figures 2(a) and 2(b) show the surfaces and cross sections of PBI workpieces machined without and with laser assistance, respectively. In the usual orthogonal machining process without laser assistance, many cracks were produced over the entire surface. In contrast, part of the surface machined using LAM was smooth, and a cross section of this area did not contain cracks. The reason for this is that the movement of the molecules in the workpiece

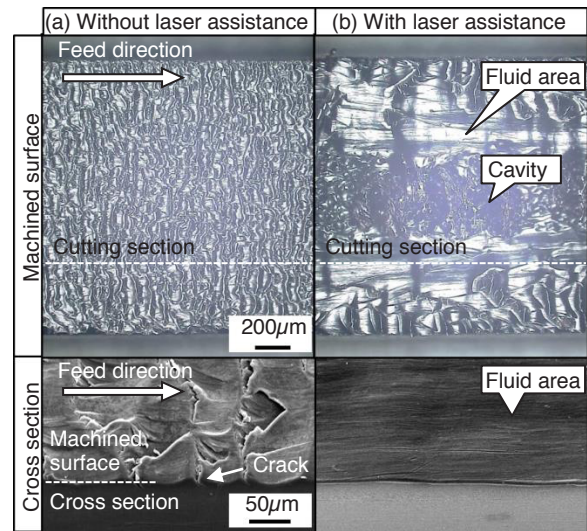


FIGURE 2. Surfaces and cross sections of PBI machined using orthogonal cutting (a) without and (b) with laser assistance.

became active as a result of the energy introduced into the workpiece, causing fluidity. Therefore, the factor that achieves this fluidity is necessary to reduce the formation of cracks, and fluid areas are defined as crack-free.

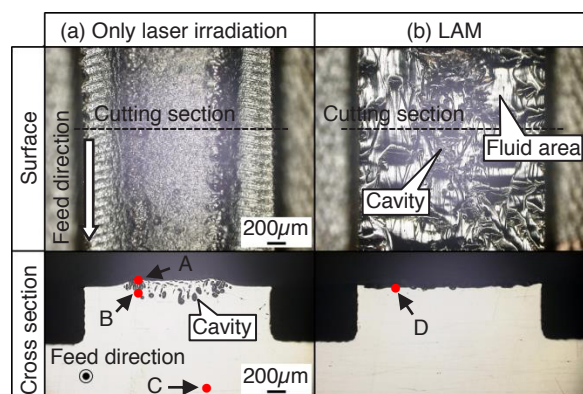
However, the center of the surface with laser assistance had many cavities, indicating the occurrence of thermal decomposition. The decomposition temperature of PBI in air is 853 K [1], and it is presumed that the temperature of the center of the cutting position exceeds 853 K. As this reason, it takes much time until machining by the tool after heating by the laser beam because feed rate is extremely slow. It produces cooling of the heated surface. Thus, the high temperatures achieved by heating using a laser are required in LAM at a slow cutting speed. However, there is a possibility that the molecular structure of PBI and its other characteristics will change. The workpiece must therefore be inspected to determine the influence of heating by the laser.

Figures 3(a) and 3(b) show surfaces which were machined using only laser irradiation and LAM. Figure 4 shows the FTIR results in microregions with diameters of 30 μm in cross section for the surfaces shown in Figure 3. In the surface machined using only laser irradiation, areas A, B, and C are defined as the microregions near the surface of the workpiece; inside the thermal

decomposition domain, where cavities develop; and in the thermally unaffected zone inside the workpiece; the FTIR results for these areas are shown in Figure 4(a)–(c), respectively. In the surface machined using LAM, area D is defined as the microregion near the surface in the fluid area; the FTIR results for this area are shown in Figure 4(d).

Area A had an absorption peak in $900\text{--}1300\text{ cm}^{-1}$, whereas area C did not. This suggests the presence of ether linkages. If an ether linkage is present in the main chain of PBI, it is possible for the glass transition temperature of PBI to decrease [7]. However, as with area C, which was not heated, area B did not show an ether linkage. This may be because benzimidazole, $\text{C}_7\text{H}_6\text{N}_2$, does not include oxygen, which is

required to construct the ether linkage. Thus, oxygen in the atmosphere was used to construct the linkage in area A, whereas in area B, which was not exposed to the air, the ether linkage could not be constructed even when the laser beam heated this area. Thus, thermally affected areas can exist only at the surface. This thermally affected surface layer will likely be removed during machining by a cutting tool and will thus not remain on the surface of finished products. In fact, area D, which was near a fluid region after LAM, did not show an absorption peak in $900\text{--}1300\text{ cm}^{-1}$; that is, there was no ether linkage in the fluid area of the workpiece machined using LAM. Thus, it was revealed that preheating using the laser beam does not produce oxygenation-induced chemical defects on the finished surface.

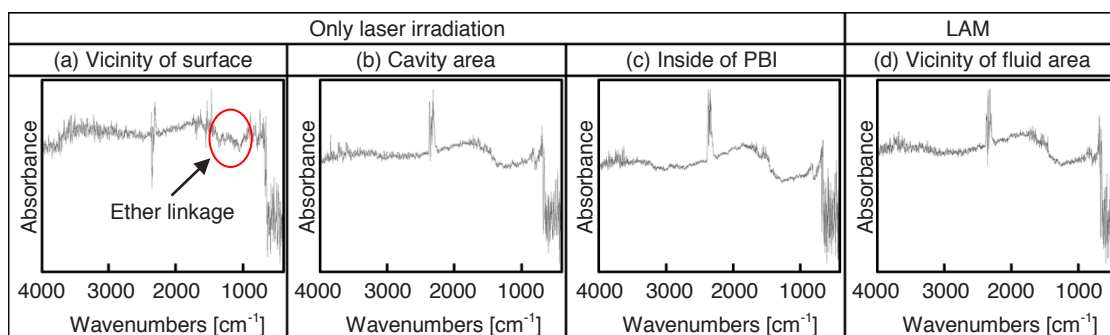


$F = 500\text{ mm/min}$, $A_d = 0.05\text{ mm}$, $D = 2\text{ mm}$, $Q = 12.5\text{ W}$

FIGURE 3. Surfaces and cross sections of PBI machined using (a) only laser irradiation and (b) LAM.

Influence of laser irradiation conditions on the fluid area of the machined surface

Obtaining the fluid area is necessary for reducing the formation of cracks; however, this area is narrower than the total machined area, as mentioned above. Because this is impractical, the laser irradiation conditions were changed to expand the fluid area. The present evaluation method considers variations in the ratio of the fluid area to the total machined area. Figure 5 shows the results under the following conditions: a laser diameter of 2 mm; laser powers of 10.2, 12.5, and 15.9 W, and distances between the laser spot and the cutting tool of 6 and 15 mm. At laser powers of 10.2 W or less, fluid areas did not form regardless of the distance between the laser spot and the tool. Therefore, many cracks formed in the machined surface under this condition, as in surfaces machined without laser



$F = 500\text{ mm/min}$, $A_d = 0.05\text{ mm}$, $D = 2\text{ mm}$, $Q = 12.5\text{ W}$

FIGURE 4. FTIR results in microregions (a) in the vicinity of the surface of the workpiece, (b) in the thermal decomposition domain, and (c) inside the workpiece under only laser irradiation and (d) in the vicinity of the fluid area under LAM.

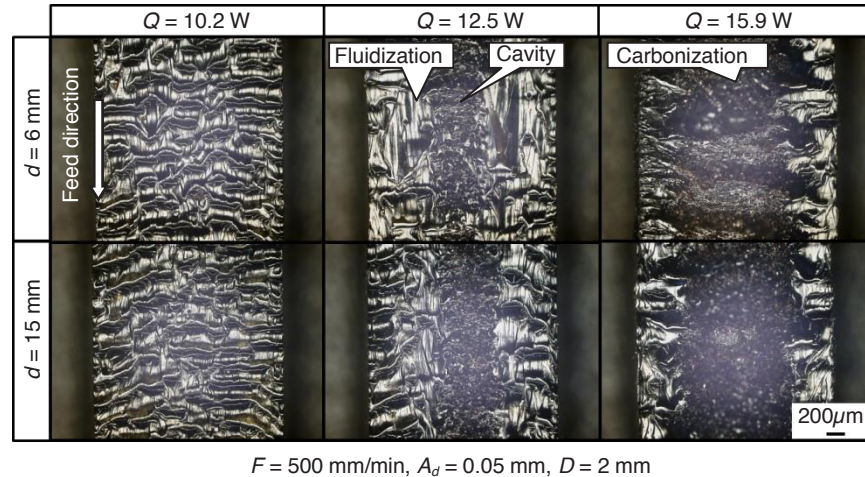


FIGURE 5. Comparison of surfaces machined using LAM under different irradiation conditions.

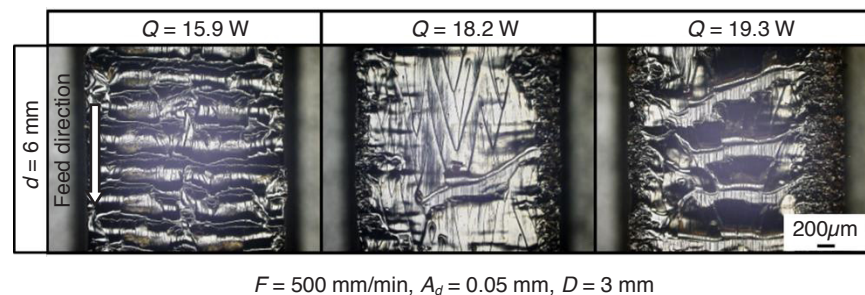


FIGURE 6. Comparison of surfaces machined using LAM under different irradiation conditions.

assistance. At a laser power of 12.5 W, even though thermal decomposition occurred in the center of the machined area, fluid areas were present on the area. However, cracks were also observed on both sides of the machined surface. At a laser power of 15.9 W, the fluid areas were very small because of the increased gasification and carbonization areas. For this reason, it was inferred that temperatures near the thermal decomposition temperature of PBI are appropriate temperatures for crack-free machining. Additionally, regarding the distance between the laser and the tool, the ratio of the fluid area to the total machined area at a distance of 6 mm was larger than at a distance of 15 mm. This indicates that the distance should be as small as possible. This is because as the time between the heating and cutting of a location on the surface of the workpiece increases, the temperature at that location at the time of cutting decreases as a result of heat dissipation in the atmosphere or thermal conduction to the inside of the material.

However, as shown in Figure 5, changing the laser power and the distance between the laser spot and the tool were not effective methods of increasing the ratio of the fluid area, as evidenced by the presence of a large number of cracks and extensive gasification. Accordingly, another experiment involving changing the laser spot diameter was carried out. Figure 6 shows the surfaces of a PBI workpiece machined using LAM under the following conditions: a laser diameter of 3 mm; laser powers of 15.9, 18.2, and 19.3 W; and a distance of 6 mm between the laser spot and the cutting tool. These results reveal larger fluid areas at a power of 18.2 W than under the conditions shown in Figure 6, though the required laser power was higher than previous condition: the laser diameter of 2 mm as a result of the decreased energy density and increased laser spot diameter. At a laser diameter of 2 mm, the temperature gradient on the heated surface of a projection of 2 mm in width will be large. In contrast, a laser spot with a larger diameter can uniformly heat the surface of the projection and enables heating to reduce

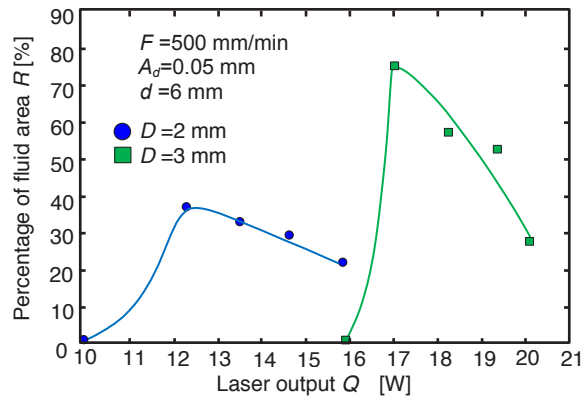


FIGURE 7. Plot of percentage of fluid area on machined surface against laser power.

the temperature gradient. For this reason, it was inferred that the proper temperature field for crack suppression increased in size. Figure 7 shows a plot of the percentage of the fluid area of the machined surface against the laser power at laser spot diameters of 2 and 3 mm. The laser diameters of 2 and 3 mm produced maximum fluid area percentages of 35% and 75%, respectively. As a result, enlarging the laser spot is an effective method of increasing the fluid area.

Temperature distribution model

Measuring the actual cutting temperature during processing is extremely important because of the effect of the softening and glass transition temperatures of PBI. However, it is difficult to measure the cutting temperature because the orthogonal cutting speed is very low and the workpiece is an insulator. These conditions create disturbances that can prevent accurate measurements. In this case, the temperature distribution was modeled using an Excel spreadsheet based on the experimental conditions, the physical properties of PBI, and

the heat conduction equation [8]. Because the temperature distribution around the laser spot stabilizes after sufficient time has passed from the initiation of heating, it was calculated under quasi-steady-state conditions. Based on the assumption that the laser spot is composed of numerous infinitesimal point heat sources, the calorific value of the spot can be regarded as the sum of the point heat capacities; the heat transferred to the workpiece was calculated in the same manner [8].

The parameters of the Excel model were calculated using the following equations:

$$T = \frac{q}{2\pi\lambda r} e^{\frac{-v(x+r)}{2a}} \quad (1)$$

$$r = \sqrt{x^2 + y^2 + z^2} \quad (2)$$

$$a = \frac{\lambda}{\rho c} \quad (3)$$

where T is the increase in the temperature of the workpiece from its initial temperature, q is the calorific value of an infinitesimal point heat source, v is the feed rate, r is the distance from the heat source, x , y , and z are the coordinate component values of the distance, a is the thermal diffusivity of the workpiece, λ is its thermal conductivity, ρ is its density, and c is its specific heat.

TABLE 2. Analysis conditions.

Thermal conductivity: λ [W/m-K]	0.41
Density: ρ [kg/m ³]	1300
Specific heat: c [J/kg-K]	930
Laser spot diameter: D [mm]	2
Laser output: Q [W]	12.5
Emittance: i [-]	0.7
Feed rate: F [mm/min]	500
Initial temperature: T [K]	298

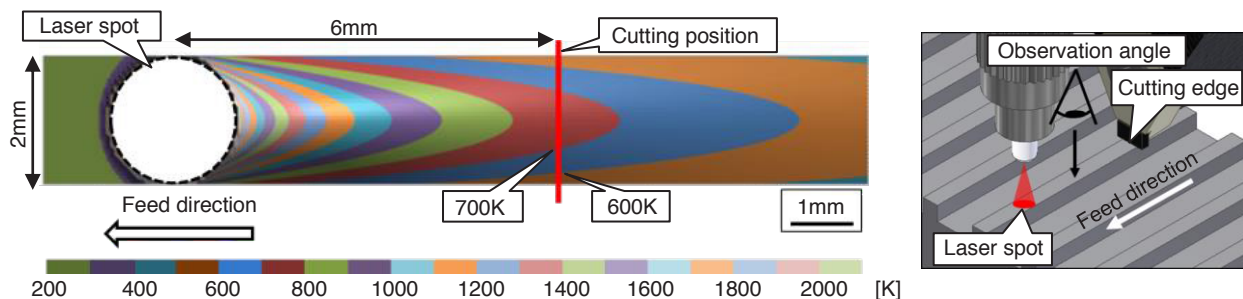


FIGURE 8. Temperature distribution near laser spot.

The analysis conditions are given in Table 2 [9]. Figure 8 shows the theoretical temperature distribution at a location on the surface of the workpiece obtained based on the calculation results. For a cutting position 6 mm from the center of the laser spot, the temperature at the center of the machined surface at this cutting position reached 700 K, which is the glass transition temperature, whereas both sides of the surface at the cutting position were less than 600 K. Assuming that the PBI begins show fluidity when the temperature at the cutting position reaches approximately 700 K, this analysis and the experimental results shown in Figure 6 are in good agreement. Therefore, the proposed calculation and analysis method can be used to predict the temperature at the time of cutting and the proper machining conditions for LAM.

However, because actual experiments could have some problems regarding decomposition, carbonization, changes to the physical properties of the workpiece, and other problems related to heating, a calculation method that includes these issues must be developed to determine accurate values.

CONCLUSION

In this study, PBI was machined using laser-assisted orthogonal cutting to suppress the formation of cracks at slow cutting speeds. The conclusions of this study are summarized as follows.

- (1) When a PBI workpiece is irradiated with a laser beam, ether linkages, which decrease the glass transition temperature, are generated on the main chain of PBI. However, ether linkages can be formed only at the surface of the workpiece; thus, the affected layer will be removed as chips during machining and will not remain on the surface of processed products.
- (2) Cracks can be suppressed under temperature conditions at which the machined surface is fluid. Cracks form at lower temperatures, and decomposition or carbonization occurs at higher temperatures.
- (3) The ratio of the fluid area to the total machined surface area increases as the distance between the laser spot and the tool decreases. This is because as the time between the heating and cutting of a location on the surface of the PBI workpiece

increases, the temperature at that location at the time of cutting decreases. However, decomposition is unavoidable. At a laser diameter of 2 mm, the maximum fluid area percentage is approximately 35%, whereas the maximum is approximately 75% at a laser diameter of 3 mm because a laser spot with a larger diameter can more uniformly heat the surface of projection and thus reduces the temperature gradient.

- (4) The temperature distribution of the heated area was calculated from the experimental conditions, the physical properties of PBI, and the heat conduction equation in a quasi-steady state.

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CUTTING CHARACTERISTICS IN TURNING OF Cr-Mo-V FORGED STEEL

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INTRODUCTION

Cr-Mo-V forged steel is a material suitable for mechanical parts used in high-temperature and high-pressure environments, such as turbine rotors and turbine valves in power plants [1, 2]. In general, parts made of Cr-Mo-V forged steel are manufactured by cutting the workpiece in the final stage of the manufacturing process. Thus, it is important to understand the cutting characteristics of Cr-Mo-V forged steel to efficiently produce parts with high accuracy. However, few studies have been carried out to investigate cutting the characteristics of Cr-Mo-V forged steel.

In the present study, we experimentally investigated the cutting characteristics in the turning of Cr-Mo-V forged steel. Cylinders made of Cr-Mo-V forged steel were turned by a computer numerically controlled (CNC) lathe under various conditions using tungsten carbide (WC) and cermet tools, and the resulting tool wear was examined. We also investigated chip formation. In this study, such cutting

characteristics were compared with those of two stainless steels used in various fields, one of which was a difficult-to-machine material.

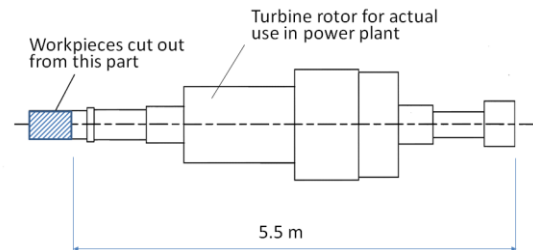


FIGURE 1. Blank of turbine rotor for actual use in power plant, from which workpieces in this study were cut out.

WORKPIECE MATERIAL

The workpieces used in this study were cut out from the blank of a turbine rotor made from Cr-Mo-V steel. The blank was made by casting followed by forging by Pacific Steel Mfg. Co., Ltd., Japan. Figure 1 shows a schematic drawing of the blank. The unhatched area of the

TABLE 1. Chemical components of Cr-Mo-V forged steel [wt%].

C	Si	Mn	P	S	Ni	Cr	Mo	V
0.31	0.08	0.78	0.007	0.002	0.41	1.10	1.13	0.22

TABLE 2. Physical and mechanical properties of workpieces.

	Hardness HV	Tensile strength	Elongation	Density	Thermal conductivity	Specific heat
	[-]	[N/mm ²]	[%]	[g/mm ³]	[W/m deg]	[x10 ³ J/kg deg]
Cr-Mo-V steel	200	795	19	7.8	45 (100 °C) 46 (200 °C) 35 (600 °C)	0.55 (260 °C) 0.63 (570 °C)
Stainless steel SUS304	200	520	40	7.9	17 (200 °C) 24 (600 °C)	0.53 (200 °C) 0.65 (600 °C)
Stainless steel SUS303	200	520	40	7.9	16 (100 °C)	0.50 (100 °C)

blank was used as an actual turbine rotor. The workpieces were produced from the hatched area on the left. It was thus assumed that the material properties of the workpieces used in our experiment were the same as those of the actual turbine rotor.

In this study, we machined the stainless steels SUS304 and SUS303, as defined by Japan Industrial Standards, for comparison. The chemical components and the physical and mechanical properties of the Cr-Mo-V forged steel are shown in Tables 1 and 2, respectively. In Table 2, the properties of the stainless steels are also shown.

TOOL WEAR

Machining of Cr-Mo-V forged steel with using and tungsten carbide tools

We investigated the tool wear generated by turning at various cutting speeds using WC and cermet tools. A CNC lathe (Okuma Corporation, SPACE TURN LB3000EX) was used for cutting. The tools had a chip breaker, and the radius of the tip of each tool was 0.4 mm. The workpieces were prepared as cylinders with 50 mm diameter and 20 mm thickness. The depth of cut and feed rate were 0.3 mm and 0.08 mm/rev, respectively, and the cutting distance was 400 m. No coolant was used during cutting. After the turning, the tool wear of a flank face was evaluated with an optical microscope.

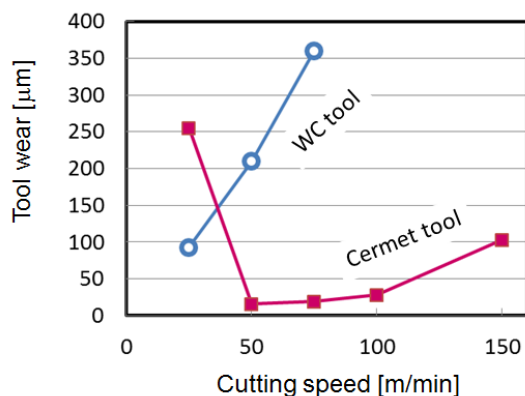


FIGURE 2. Relationship between cutting speed and tool wear for WC and cermet tools.

Figure 2 shows the experimentally obtained relationship between the cutting speed and tool wear. The tool wear for the WC tool is larger than that for the cermet tool. For the WC tool, the tool wear rapidly increased with increasing cutting speed. On the other hand, for the cermet tool, the tool wear was large at a low cutting speed of below 50 m/min and at a high cutting speed of above 100 m/min. The tool wear occurring at a low cutting speed and high cutting speed may have resulted from adhesive wear and the increase in temperature of the tool, respectively. These findings indicate that the cermet tool is superior to the WC tool and should be used at a cutting speed within the range of 50-100 m/min for the turning of Cr-Mo-V forged steel.

Machining of Cr-Mo-V steel and stainless steels

Next, we compared the tool wear for the machining of Cr-Mo-V steel with that for stainless steels SUS304 and SUS303. SUS 304 was chosen because it is considered to be a difficult-to-machine material owing to its low thermal conductivity, whereas SUS303 is a free-machining stainless steel owing to the addition of sulfur and phosphorus.

The experimental results in the previous section indicate that the cermet tool is more suitable for cutting Cr-Mo-V forged steel than the WC tool in terms of tool wear. Moreover, it should be used at a cutting speed of 50-100 mm/min. Thus, in this study, we cut cylinders made of the Cr-Mo-V forged steel and the stainless steels using the cermet tool with a cutting speed of 100 mm/min to evaluate tool wear. Then, we investigated the effect of the cutting distance on the tool wear of a flank face.

Figure 3 shows the experimental results. The tool wear for the Cr-Mo-V forged steel was greater larger than that for the stainless steels. Previous studies showed that the cutting force and cutting temperature are functions of the tool wear, where a higher cutting temperature and larger cutting force cause a larger amount of tool wear [3, 4]. As shown in Table 2, the thermal conductivity of the Cr-Mo-V forged steel is higher than that of the stainless steels. On the other hand, the tensile strength of the Cr-Mo-V forged steel is higher than that of the stainless steels, although their hardnesses are the same. Considering the thermal conductivity and tensile strength, the reason why the tool wear for the

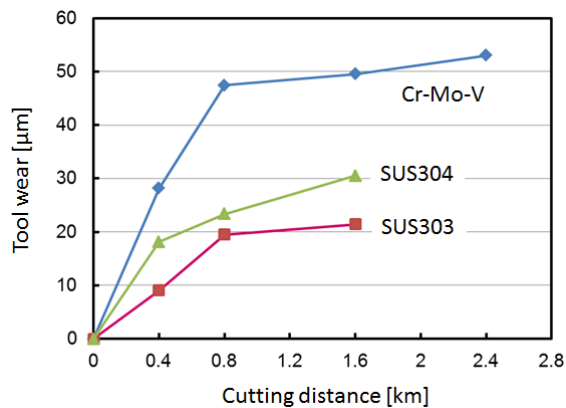


FIGURE 3. Change in tool wear with cutting distance for different workpiece materials.

Cr-Mo-V forged steel was higher than that for the stainless steels is probably that the cutting force for the Cr-Mo-V forged steel is higher than that for the stainless steels.

To confirm our hypothesis about the tool wear of the Cr-Mo-V forged steel, we measured the cutting force during turning by a cutting dynamometer. In this experiment, we used a manual lathe because our cutting dynamometer did not fit the CNC lathe used in the above experiment. The cutting speed and feed rate were set to 92 m/min and 0.16 mm/rev, respectively, which were as similar values as

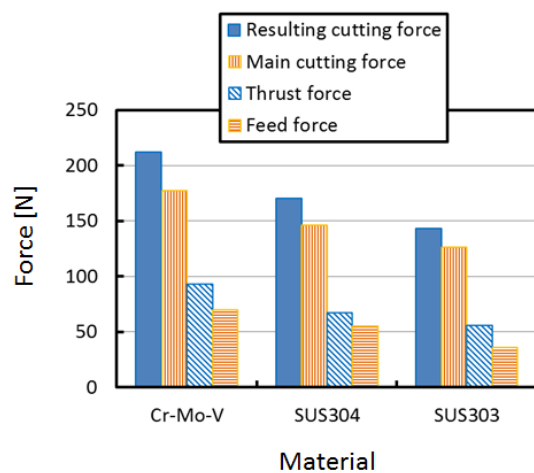


FIGURE 4. Cutting forces for different workpiece materials.

possible to those for the CNC lathe. The depth of cut was 0.3 mm, which was the same as that for the CNC lathe.

Figure 4 shows the experimental results of the cutting forces for the Cr-Mo-V forged steel and the stainless steels. The cutting force for the Cr-Mo-V forged steel was larger than that for the stainless steels. These experimental results are consistent with our hypothesis regarding the higher tool wear of the Cr-Mo-V forged steel.

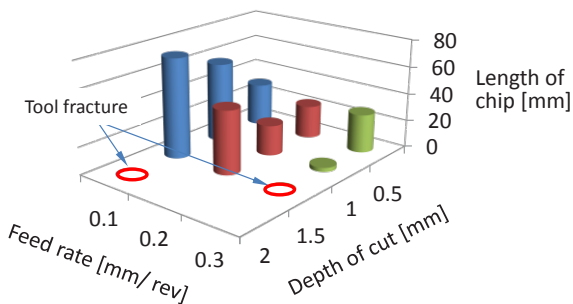
CHIP FORMATION

The characteristics of chip formation are an important factor in turning. The formation of continuous chips obstructs the operation of a turning machine in some cases. Thus, we investigated chip formation for the Cr-Mo-V forged steel in the turning process. We measured the lengths of chips generated by turning under various cutting conditions, such as the depth of cut and the feed rate of the tool. Workpieces were turned by the CNC lathe with a cermet tool used in the above experiment. The cutting speed was 100 m/min, and the stainless steels SUS304 and SUS303 were also processed for comparison.

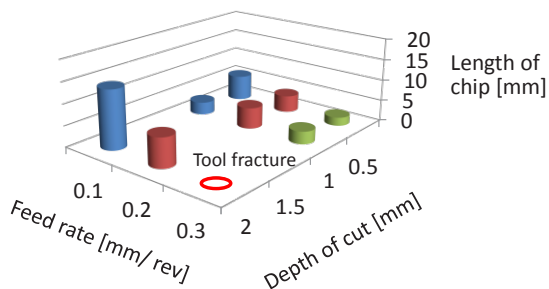
Figure 5 shows the experimental results of the chip lengths for the Cr-Mo-V forged steel and the stainless steels under various cutting conditions. The shape of the resultant chips for the Cr-Mo-V forged steel was helical in every case. In this experiment, the turning operation was not obstructed by the resulting chips. As shown in Fig. 5, the length of each chip for the Cr-Mo-V forged steel was longer than that for the stainless steels, although the elongation of the Cr-Mo-V forged steel is the lowest among the materials listed in Table 2. To understand the experimental results in more detail, the properties of each material in the high-temperature and high-pressure environments where the cutting occurs should be taken into account. For instance, the Cr-Mo-V forged steel was developed for use in high-temperature and high-pressure environments, therefore it has high creep strength in such environments, which may affect its cutting properties.

Regarding the chips of the stainless steels, the chip length for SUS303 was less than that for SUS304, which is considered to be due to the materials, such as sulfur and phosphorus, added to SUS303 to achieve high machinability.

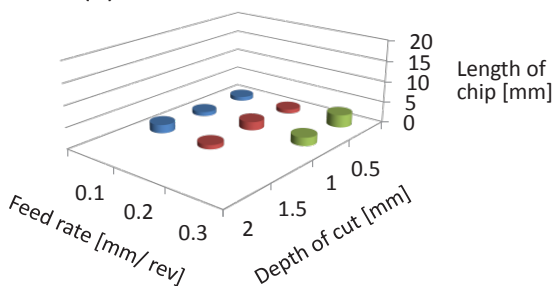
In addition, as shown in Fig. 5, the tool used in the cutting most easily fractured in the case of the Cr-Mo-V forged steel. This is considered to be due to the large cutting force for the Cr-Mo-V forged steel as shown in Fig. 4.



(a) Cr-Mo-V forged steel



(b) Stainless steel SUS304



(c) Stainless steel SUS303

FIGURE 5. Chip formation for different workpiece materials.

CONCLUSIONS

Cylinders made of Cr-Mo-V forged steel were turned by a CNC lathe under various conditions to investigate its cutting characteristics. For comparison, cylinders made of stainless steels were also cut. It was found that the wear of a cermet tool was less than that of a WC tool and that it became small at a cutting speed of 50-100 m/min. Moreover, the tool wear for the cutting of Cr-Mo-V forged steel was larger than that for the stainless steels. The measurement of the cutting force revealed that the Cr-Mo-V forged steel required a larger cutting force than the stainless steels, which may have results in the larger tool wear for the cutting of the Cr-Mo-V forged steel. We investigated chip formation for the Cr-Mo-V forged steel in the turning process, which showed that the length of chips of the Cr-Mo-V forged steel was greater than that of the stainless steels, although the elongation of the Cr-Mo-V forged steel is lower than that of the stainless steels.

ACKNOWLEDGEMENTS

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Laser Micro-structuring Electroplated Coarse-Grained Diamond Wheels for Precision Grinding on Optical Glasses

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INTRODUCTION

The conditioned coarse-grained diamond grinding wheels are able to achieve the identical surface roughness, higher form accuracy and larger material removal rate on optical glasses due to their better wear resistance ability, longer grinding wheel life and bigger chip space when compared to the traditional fine-grained diamond wheels^[1-3]. However, deeper subsurface damage could be inevitably introduced due to the larger flat tops of the coarse diamond grains^[4-6].

In order to overcome this problem and achieve high-efficiency precision grinding of optical glasses, a new deterministic grinding process with surface laser micro-structured electroplated coarse-grained diamond grinding wheels was introduced. The previous work indicated that the regularly micro-grooves (groove width from several microns to tens of microns) on the coarse-grained diamond grains' tops are able to reduce the contact area between work-piece and the flat tops of the coarse diamond grains. Therefore, the normal force and tangential friction force in grinding process can be decreased considerably, resulting in the improvement of the subsurface damage^[7].

In this paper, a novel conditioning method was developed for electroplated coarse-grained diamond grinding wheels. And then, the defined patterns were micro-structured on diamond surfaces successfully by means of a sub-nanosecond pulsed laser. Furthermore, three surface micro-structured grinding wheels with different abrasive grain sizes were prepared. At last, the grinding experiments were performed to make sure the influence of these micro-structures on grinding processes.

EXPERIMENTAL SETUPS

Grinding and conditioning setup

Figure 1 is the grinding and conditioning setup. The grinding machine tool used in this research was a surface grinder with three orthogonal translation axis (0.1 μ m position accuracy) and a hydro-static spindle. A vitrified diamond conditioning wheel was mounted on the hydro-static spindle of the grinder and an extra precision spindle was fixed on the X-axis as the grinding spindle. The electroplated coarse-grained diamond wheel was carefully mounted on the extra precision spindle. The work-piece and dynamometer were fixed on Z-axis with enough stiffness. The grinding forces were measured and recorded by the dynamometer in each grinding operation. The grinding coolant was emulsion liquid and be delivered to the grinding zone by a nozzle.

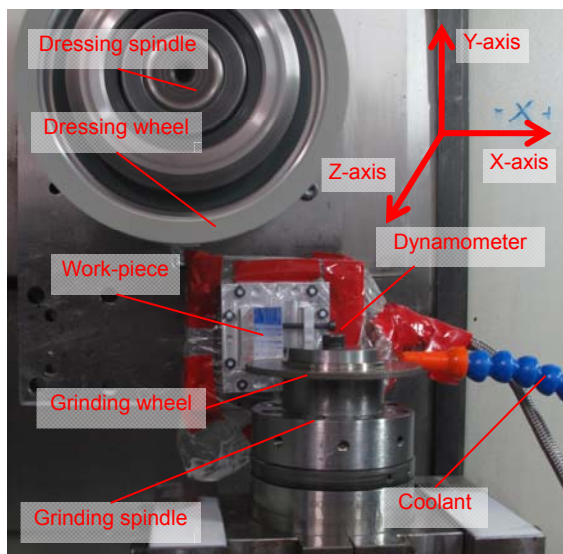


FIGURE 1. Grinding and conditioning setup

Laser micro-structuring setup

A sub-nanosecond pulsed laser of wave length 532nm, pulse width 0.8ns, pulse repetition frequency 1-5kHz, initial laser beam diameter

3mm, divergence angle 7mrad and Gaussian energy distribution was taken as the laser source for micro-structuring the diamond surface and grinding wheels. The laser micro-structuring setup, as shown in figure 2 and figure 3, was composed of laser source, mirrors, focus lens, CCD cameras, laser distance sensor and a 4-axis precision translation stage (position accuracy 0.1 μ m). The laser beam is propagated from the laser source to the focus lens and the laser power and repetition were directly adjusted in the laser source. The focus lens was a 10 $\times$ microscope objective and can focus the laser beam to less than 20 μ m at the focus plane. The CCD cameras were used to on-machine observe the machined surface and the laser distance sensor was as a reference for determine the work-piece surface position in Z-axis and help to translate the work-piece surface to the focus plane. The coordinated movement of the translation stage was able to obtain the desired micro-structuring patterns.

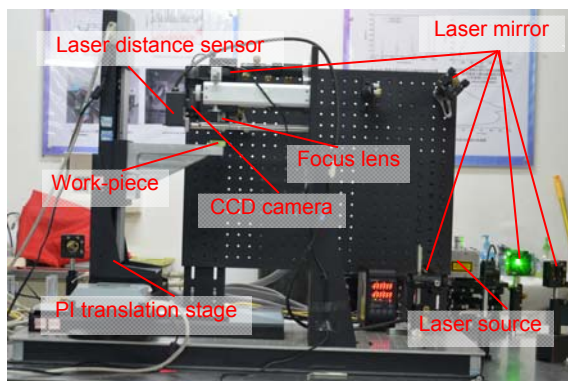


FIGURE 2. Micro-machining setup

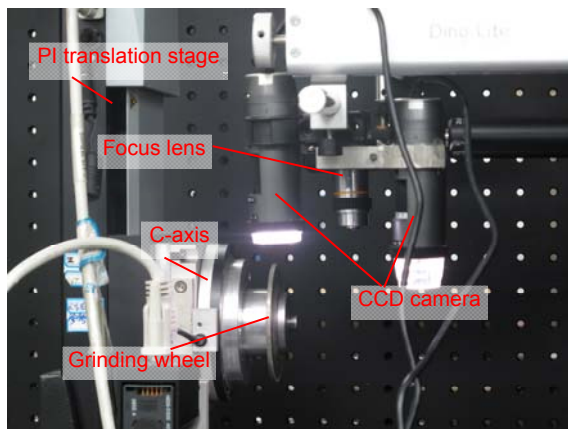


FIGURE 3. Micro-structuring setup

RESEARCH RESULTS

Conditioning electroplated coarse-grained diamond grinding wheels

The electroplated coarse-grained diamond grinding wheels were conditioned by a vitrified fine-grained diamond cup-shape conditioning wheel. In the conditioning process, the grinding wheel translated in the X-axis and in-feed in the Z-axis per pass with 1 μ m. Therefore, the higher diamond grains were more and more truncated and remains a flat top surface. Figure 4 shows the abrasive grains of grinding wheel with grain size of 70# (D213) before and after conditioned. It is clear that the abrasive grains possess random shapes, protrusion height, orientations and almost no flat top surface before conditioning. According to the research results of Brinksmeier and Zhao^[1-5], the original topography of electroplated coarse-grained diamond grinding wheel is not suitable for precision grinding hard and brittle materials. After a total dressing depth of about 180 μ m, the abrasive grains are truncated to the same protrusion height and form a flat top surface. The effective cutting edge length per abrasive grain is also sharply increased after well-conditioned.

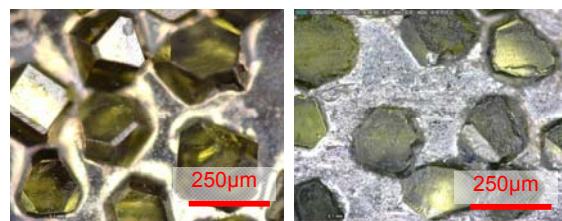
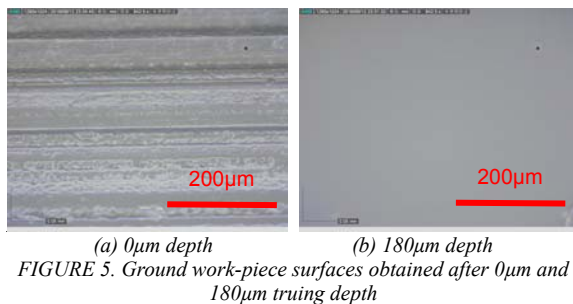


FIGURE 4. The diamond grain(70#) before and after conditioning

Subsequently, two optical glass BK7 work-piece were ground after the conditioning process. As described in Figure 5(a), the work-piece surface obtained before conditioning exists obvious groove profiles and the large white areas in the work-piece surface means the material were removed mainly in brittle regime although some ductile regime removed areas were also can be observed. The big initial protrusion height error of the diamond grits in the axis direction of the grinding wheel resulted in the serious groove profiles while the initial

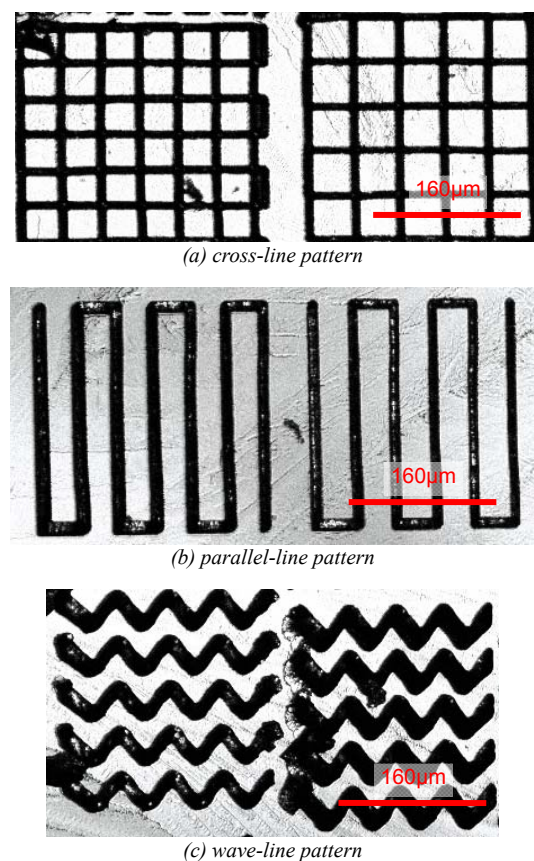
circular run-out and inefficient effective abrasive grains and cutting length per abrasive grain cause the material was mainly removed in brittle regime. Figure 5(b) is the work-piece surface obtained after 180 μm truing depth, there is almost no groove profile, even no grinding traces can be observed. That means all the ground surface was formed in ductile regime. The surface roughness of surface obtained after 180 μm truing depth measured by atom force microscopy (AFM) is Ra 3.6nm and Ra 1.1nm in the vertical and parallel direction to the grinding direction, respectively. Therefore, precision grinding optical glass material with nanometer scaled surface roughness is possible with well conditioned electroplated coarse-grained diamond grinding wheels.



Laser micro- structuring on diamond

For effective and efficient micro-structuring conditioned coarse-grained diamond grinding wheels with the defined pattern and structuring ration (the proportion of the area remained after micro-structuring), to determine the laser parameters such as scanning speed, average power, repetition frequency and focus position and the strategy for micro-structuring operations beforehand is very important and meaningful. Based on the micro-machining experimental research results, the parameters for micro-structuring diamond surface is scanning speed 0.25mm/s, average power 80mW, repetition frequency 5kHz and off-focus quantity 0 μm and the machined groove width is about 10 μm . With the determined laser machining parameters, three different patterns include cross-line array, parallel-line array and wave-line

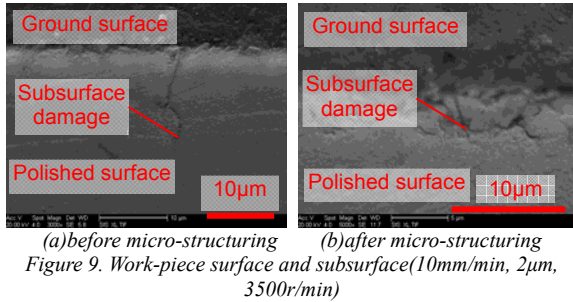
array were machined on single diamond grit as shown in Figure 6. There is something need to be explained that the groove width of the wave-line array was bigger than others patterns due to an extra scanning operation was performed with a deviation of half and one groove width, respectively. The micro-structuring ratio of the patterns in Figure 6 is 50-75% and is easy to be determined with image processing method. Furthermore, the groove edge of the patterns is sharp and no serious crack can be discovered. Consequently, it is safe to conclude that micro-structuring diamond surface with a sub-nanosecond pulsed laser is possible.



Micro-structuring grinding wheels

Based on the above experimental results and the defined structuring ratio 66.7% in advance, four surface laser micro-structured conditioned coarse-grained diamond grinding wheels with different abrasive grain sizes and patterns were prepared as shown in Figure 7. The groove width of these patterns was 30 μm .

with angle polishing method. The subsurface damage depth decreased from 12 μ m to 4 μ m.



CONCLUSIONS

In this research, a novel grinding process is introduced and some conclusion can be concluded as follows:

(1)Electroplated coarse-grained diamond grinding wheel can be effectively conditioned by vitrified diamond wheel. The optical glass surface with roughness Ra of less than 5nm was obtained.

(2)Sub-nanosecond pulsed laser is suitable for micro-structuring diamond wheel surface. Micro-structured diamond grain surface with sharp groove edge and the defined patterns without serious crack could be obtained.

(3)Although the ground surface roughness was increased by the micro-structured grinding wheel, the grinding force was reduced by about 35%-74% and the subsurface damage depth was restrained obviously.

ACKNOWLEDGE

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A STUDY ON THE TURNING OF TITANIUM ALLOY Ti-6Al-4V WITH ULTRASONIC LUBRICATION

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INSTRUCTIONS

The titanium alloy Ti-6Al-4V has been widely applied in the aerospace, medical, chemical and other industries in recent years due to its unique mechanical and chemical properties such as high strength-to-weight ratio, low density and outstanding corrosion resistance. Unfortunately, inherently poor thermal conductivity, low elasticity modulus and high chemical activity lead to poor machinability in conventional turning (CT) process.

To tackle this problem, ultrasonic vibrations were implemented to assist the CT process and several satisfactory achievements had been obtained on cutting force decrease and tool damage relief [1, 2, 3]. Nevertheless, in most of the existing researches, the superimposed directions of ultrasonic vibrations contained the primary cutting direction, in which the cutting speed couldn't exceed a certain critical value otherwise the benefits of ultrasonic vibration would be prevented by the continuous contact between workpiece and cutting tool, eventually limiting the material removal rate. Therefore, to elevate the cutting speed and increase the material removal rate, one of the present authors proposed an alternative elliptical ultrasonic assisted turning (EUAT) technique in his previous work [4], in which no vibration is applied in the cutting speed direction but superimposed in both radial and feed/axial directions in the base plane, and experimentally validated the proposed method in the turning of Ti-6Al-4V and Inconel 718 [5]. This method benefits from the reduction of the friction coefficient between the tool and the workpiece for the sake of the ultrasonic lubrication effect [6].

However, the material removal mechanism and the machining characteristics in the turning process of difficult-to-machine materials such as Ti-6Al-4V by the proposed EUAT method are still poorly understood and considerable research work remains to be carried out. In this paper, for comprehending, enhancing and expanding the application of EUAT in the high-productivity machining of difficult-to-machine materials, the fundamental turning characteristics of Ti-6Al-4V is studied by experimentally investigating the effects of cutting parameters (cutting speed, feed rate and depth of cut) on the machining characteristics in terms of cutting force, surface morphology/roughness and chip formation under the ultrasonic lubrication in the comparison with the conventional turning without the ultrasonication.

MACHINING PRINCIPLE AND EXPERIMENTAL SETUP

Figure 1 illustrates the machining principle of the proposed EUAT method. An XYZ-coordinate system is applied which obeys the right hand rule and keeps compatible with the common Cartesian coordinate system of CNC lathes; the X, Y and Z axes correspond to normal, tangential and feed directions, respectively. The workpiece rotates around its own axis (i.e., Z axis) at a speed of n_w to provide the primary motion, while the centroid of cutting tool performs left-hand longitudinal feed at a rate of f_r from Z+ direction to Z- direction at a cutting depth of a_p as the secondary motion. In addition to the conventional parallel turning motion, the cutting tool simultaneously executes forced high-frequency harmonic vibration along the X and Z axes at amplitudes of A_x and A_z with a certain phase shift which results in an elliptical motion of

the cutting tool with a phase offset φ . Due to the composition of the abovementioned motions, the cutting tool tip motions along an elliptical locus, eventually leading to a wavy machined work-surface as shown in the bottom-right corner of Figure 1.

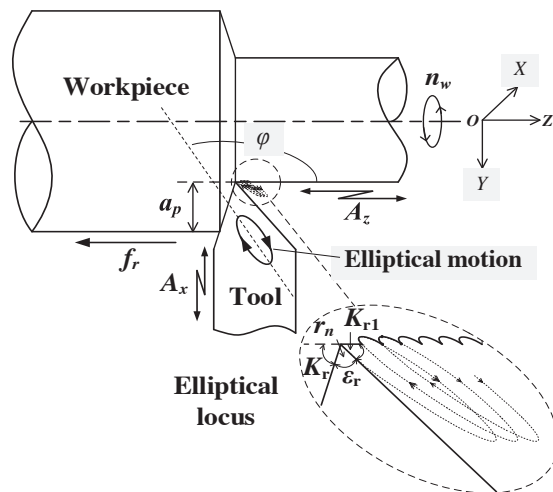


FIGURE 1. Machining principle of the EUAT technique

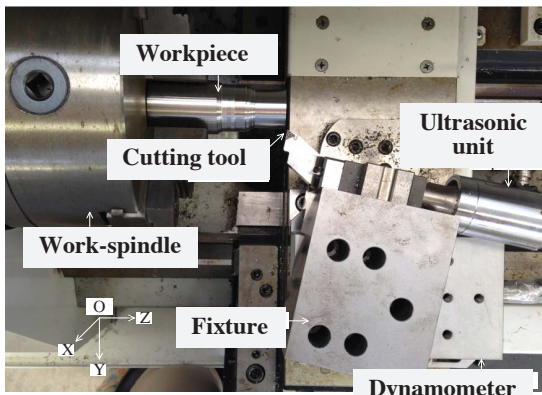


FIGURE 2. A picture of the experimental setup

Figure 2 shows a picture of the experimental setup established in-house for realizing the above-described machining principle. A commercially available ultrasonic cutting unit (UR40-A1 by Takesho Co., Ltd., Japan) was installed onto a commercial CNC lathe (TAC-360 by Takizawa Machine Tools Co., Ltd., Japan) through a rigid rectangular fixture. The ultrasonic cutting unit contained two pairs of orthogonal synchronous piezoelectric (PZT) actuators inside its ultrasonic transducer to form an elliptical vibration trajectory by an integrated resonant beam structure. A triangular carbide turning insert (TPGH080204R-FS-HTi10 by Mitsubishi Materials Co., Ltd., Japan) whose

high temperature resistance and hot hardness were well suited for the machining of Ti-6Al-4V was mounted at the end of the tool holder of the ultrasonic cutting unit and ensured to lie in the same horizontal plane with the axis of cylindrical workpiece at right angles and vibrate elliptically in the base plane and perpendicularly to the primary cutting direction. A three-component PZT dynamometer (9257B by Kistler Co., Ltd., Switzerland) was fixed below the tool holder. A tabletop stylus-type profilometer (Form Talysurf Intra; Taylor Hobson Inc., U.K.) was used to measure the work-surface roughness. A color three-dimensional (3D) laser microscope (VK-8710, KEYENCE Co., Ltd., Japan) was used to examine the work-surface morphology. Moreover, a 3D scanning electron microscope (ERA-8900 by ELIONIX Co., Ltd., Japan) was used to observe the formed chips and measure their sizes.

The ultrasonic vibration amplitude of the tool is adjusted by altering the electric voltage applied to the PZT actuators within the cutting unit from low to high level. To investigate the effect of vibration amplitude on machining characteristics, the ultrasonic vibrations of the cutting tool in the X and Z axes were simultaneously measured before turning experiments using two orthogonal Laser Doppler vibrometers (LV-1610 by Ono Sokki Co., Ltd., Japan) and the measurement results demonstrated that the tool tip is actually capable of elliptically vibrating at an ultrasonic frequency of $f_v=40.9\text{kHz}$ with a phase offset of $\varphi=168^\circ$. The vibration amplitude in X and Z-axes were up to $A_x=2.03\mu\text{m}$ and $A_z=1.92\mu\text{m}$, respectively. Turning experiments with and without ultrasonic vibration were conducted under the experimental conditions as shown in Table 1. Obviously, the lowest cutting speed $V_c=40\text{m/min}$ in this work exceeded the critical speed in traditional EUAT, i.e., $2\pi f_v A_z=15\text{m/min}$ (when $f_v=40.9\text{kHz}$, $A_z=0.98\mu\text{m}$). The workpiece used was a rod-shaped Ti-6Al-4V specimen of 35mm in diameter and 100mm in length. In experiments, the single factor analysis method was applied; that is, a single parameter changed in the given range while the rest ones remained constant during turning.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Cutting Forces

Figure 3 shows the effects of the ultrasonic vibration amplitude (A_x , A_z) on the three compo-

TABLE 1. Experimental conditions.

Workpiece	Ti-6Al-4V, $\phi 80$ mm $\times$ L120 mm
Cutting tool	Material: Uncoated tungsten carbide
	Rake angle $\alpha=0^\circ$
	Clearance angle $\gamma=11^\circ$
	Cutting edge angle $K_r=78^\circ$
	Minor cutting edge angle $K_{r1}=42^\circ$ ($\epsilon_r=60^\circ$)
	Nose radius $r_n=0.4$ mm
Cutting parameters	Cutting speed $V_c=44$ -132m/min
	Feed rate $f_r=0.05$ -0.25mm/rev
	Depth of cut $a_p=0.1$ -0.9mm
Ultrasonic vibration	Frequency $f_u=40.9$ kHz
	Amplitudes (A_x, A_z) = (1.12,0.98), (1.55,1.44), (2.03,1.92) (μm)
	Phase offset $\phi=168^\circ$
Coolant	Without (Dry turning)

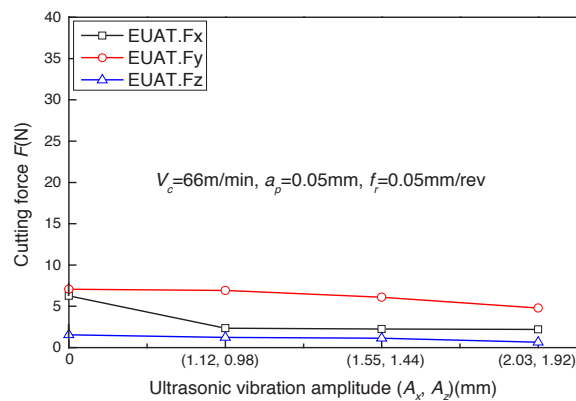


FIGURE 3. Cutting forces vs. vibration amplitude

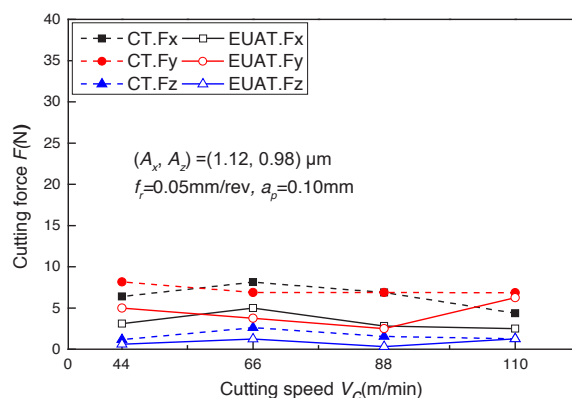


FIGURE 4. Cutting forces vs. cutting speed in CT and EUAT

nents of the cutting force in directions of tangential (X), normal (Y) and feed (Z), F_x , F_y and F_z , obtained at $V_c=66\text{m/min}$, $f_r=0.05\text{mm/rev}$,

and $a_p=0.05\text{mm}$. It can be seen that in CT where $(A_x, A_z)=(0, 0)$, the cutting force components were $F_x=6.1\text{N}$, $F_y=7\text{N}$ and $F_z=1.8\text{N}$, and they decreased to $F_x=2.3\text{N}$, $F_y=6.8\text{N}$ and $F_z=1.7\text{N}$, by 62%, 3% and 6%, respectively, once the tool has elliptically vibrated at the amplitude of $(A_x, A_z)=(1.12\mu\text{m}, 0.98\mu\text{m})$. Then, as the amplitude increased, the F_y and F_z continued to decrease whilst the F_x decreased little. Given that the cutting speed $V_c=66\text{m/min}$ was considerably over the critical value of 15m/min in traditional EUAT, the turning process can not benefit from the interruption behavior between the tool rake face and the chip as can be seen in the traditional EUAT, hence the decrease in cutting force in this work is predominantly attributed to the ultrasonic lubrication.

Regarding the effect of the elliptic ultrasonic vibration on the cutting force at different cutting speeds, the experimental results obtained at $f_r=0.05\text{mm/rev}$ and $a_p=0.1\text{mm}$ in CT ($(A_x, A_z)=(0\mu\text{m}, 0\mu\text{m})$) and EUAT ($(A_x, A_z)=(1.12\mu\text{m}, 0.98\mu\text{m})$) are as shown in Figure 4. It can be found that regardless of the cutting speed, all the force components in EUAT were greatly smaller than those in CT, and in CT they tend to decrease as the cutting speed increases, whereas in EUAT they are nearly independent of the cutting speed.

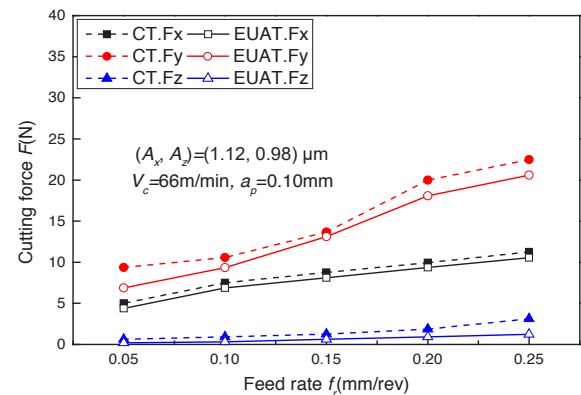


FIGURE 5. Cutting forces vs. feed rate in CT and EUAT

As for the effect of the feed rate f_r on the cutting force, Figure 5 shows the results attained at $V_c=66\text{m/min}$ and $a_p=0.1\text{mm}$ in CT and in EUAT ($A_x=2.03\mu\text{m}$, $A_z=1.92\mu\text{m}$). Evidently, either in CT or in EUAT all the three components of cutting force increased with the increase of f_r linearly and all the components in EUAT were smaller than those in CT regardless of the value of f_r . However, it is also figured out that the force

decrease percentage owing to the elliptic ultrasonic vibration became small gradually as the f_r increases. This might be because the ratio A_z/f_r , i.e., the relative impact of ultrasonic vibration in the feed direction, decreased with the increase of the feed rate.

Figure 6 shows the effect of the depth of cut a_p on the cutting force in CT and in EUAT ($(A_x, A_z)=(1.12\mu\text{m}, 0.98\mu\text{m})$) at $V_c=66\text{m/min}$ and $f_r=0.05\text{mm/rev}$. Obviously, all the three force components increased with the increase of the a_p in both EUAT and CT process, which is probably attributed to the increase of the workpiece deformation. Similarly, the force decrease percentage owing to the ultrasonication was gradually reduced as the a_p increased. This might be because the ratio A_x/a_p , i.e., the relative impact of ultrasonic vibration in the normal direction, decreased with the increase of the depth of cut.

Work-surface Morphology and Roughness

The morphologies of typical machined work-surfaces by CT and EUAT are compared in Figure 7. In CT (Figure 7(a)), parallel distributed cutting marks were formed on the work-surface, whilst in EUAT (Figure 7(b)), cusp-shaped knurls were generated on the work-surface besides the parallel distributed cutting marks. The distance between two knurls measured along the cutting direction was $26.87\mu\text{m}$ which was well consistent with the result $26.89\mu\text{m}$ calculated by the formula of $V_c/(60f_r)$, indicating these knurls were generated by the ultrasonic vibration of tool in the Y-direction. Further, built-up edges (BUE) seemed more apparent in CT than EUAT. However, the work-surface roughness of $Ra0.26\mu\text{m}$ in EUAT was a little larger than that of $Ra0.24\mu\text{m}$ in CT. This might be attributed to the increase in the relative height between the cusp-shaped knurls and the parallel distributed cutting marks owing to the ultrasonic vibration in the Y-direction normal to the work-surface.

The effects of the ultrasonic vibration amplitude and the turning parameters on the work-surface roughness are plotted in Figures 8-11, respectively. First paying the attention to the effect of the vibration amplitude, it can be seen from Figure 8 that when the cutting speed, the feed rate and the depth of cut were kept constant at $V_c=66\text{m/min}$, $f_r=0.05\text{mm/rev}$ and $a_p=0.05\text{mm}$, respectively, both the Ra and Rz increased with the increase of vibration amplitude owing to the increase in the knurl

height caused by the vibration in the direction normal to the work-surface.

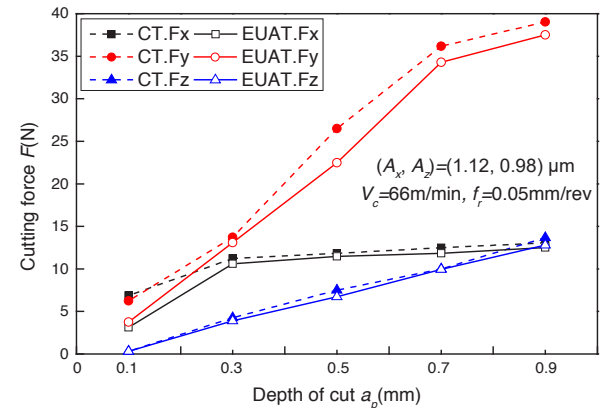
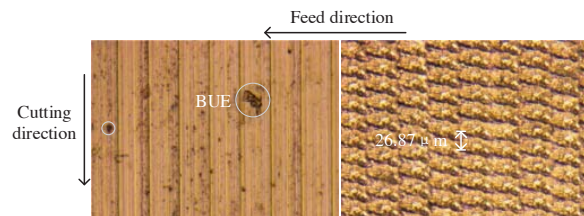


FIGURE 6. Cutting forces vs. depth of cut in CT and EUAT



(a) CT ($Ra0.24\mu\text{m}$) (b) EUAT ($Ra0.26\mu\text{m}$)
FIGURE 7. Work-surface morphologies in CT and EUAT

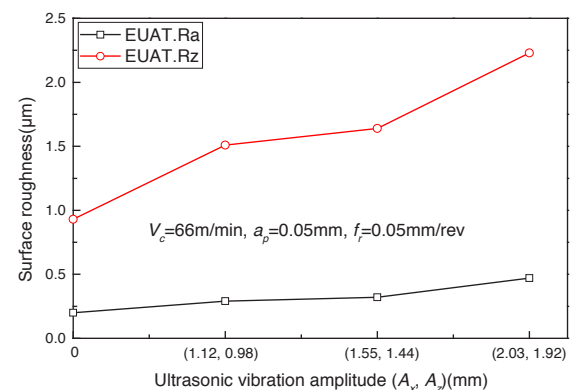


FIGURE 8. Work-surface roughness vs. vibration amplitude

Figure 9 shows the effect of the cutting speed V_c on the Ra and Rz in CT and EUAT ($(A_x, A_z)=(1.12\mu\text{m}, 0.98\mu\text{m})$) at $a_p=0.1\text{mm/min}$ and $f_r=0.05\text{mm/rev}$. It is interesting to note that both the Ra and Rz in CT tended to increase as the v gets larger whereas in EUAT they were in reduction tendencies. In particular, although at beginning the surface roughness in EUAT were larger than that in CT, once the v exceeded

66m/min both the Ra and Rz in EUAT turned to be smaller than that in CT. This indicates that the elliptic ultrasonic vibration has a positive effect on work-surface roughness in high speed turning.

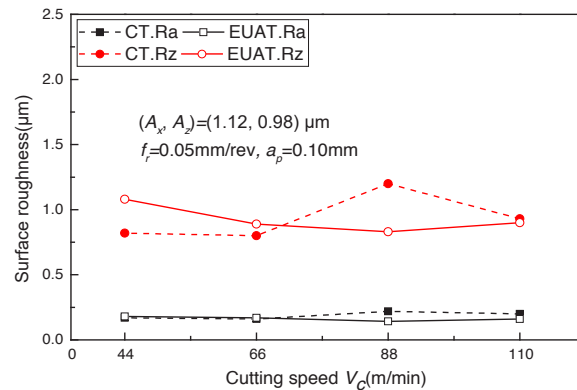


FIGURE 9. Work-surface roughness vs. cutting speed in CT and EUAT

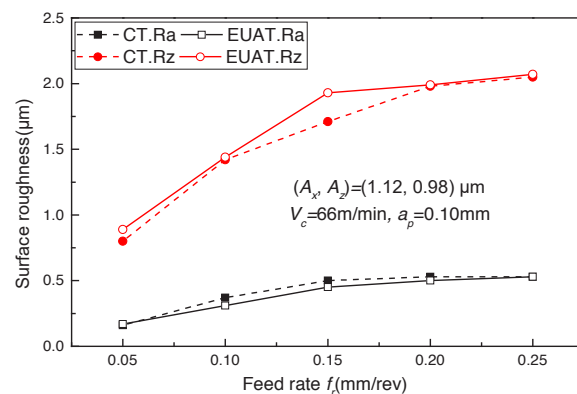


FIGURE 10. Work-surface roughness vs. feed rate in CT and EUAT

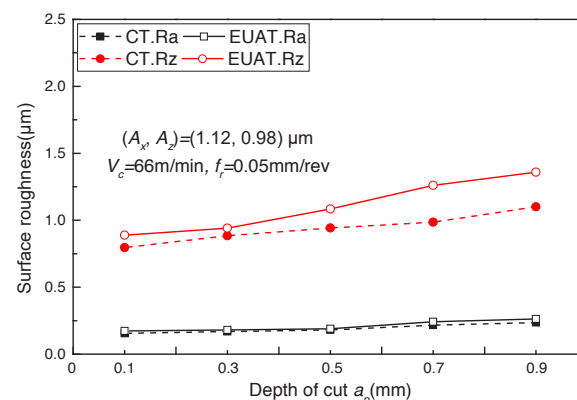


FIGURE 11. Work-surface roughness vs. depth of cut in CT and EUAT

Regarding the effect of the feed rate on the surface roughness, it can be seen from Figure 10 that either in CT or in EUAT, higher feed rate results in larger work-surface roughness. Shifting the attention to the effect of the ultrasonic vibration reveals that when the feed rate is around 0.15mm/rev, the ultrasonic vibration deteriorated the surface roughness somewhat whereas once the feed rate increased to over 0.2mm/rev, the surface roughness deterioration caused by the ultrasonic vibration is hardly observed.

From Figure 11, it can be seen that either in CT or in EUAT both the Ra and Rz increased with the increasing depth of cut and both the Ra and Rz in EUAT were larger than those in CT regardless of the depth of cut.

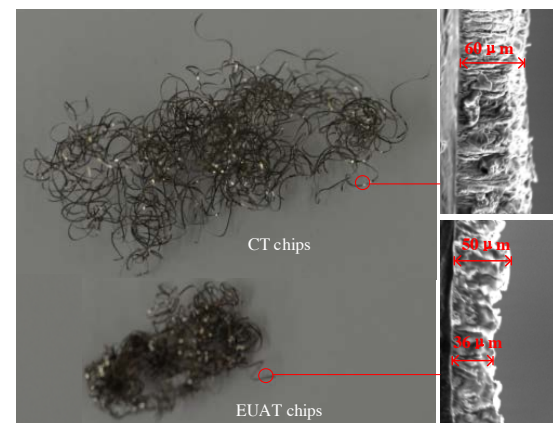


FIGURE 12. Cutting chips formed in CT and EUAT

Cutting Chips

The tool insert with groove-type chip breaker was employed during the experiments. Photographs of formed chips are shown on the left side of Figure 12, while the SEM images of single chips obtained in CT and EUAT are shown on the right side of Figure 12. It can be observed that the chips formed in CT were continuous ribbon type, which tended to be more accumulative and entangled with the tool and uncut workpiece than those in EUAT. By contrast, the snarled washer-type chips with smaller curled radii were formed in EUAT, which were easier to be broken and became shorter than those in CT, resulting in the improvement of machinability. Moreover, the minimum chip thickness in CT was 60 μm , even bigger than the maximum chip thickness in EUAT which was 50 μm . The chip thickness in EUAT changed more dramatically than that in CT due to the

dynamic alteration of the real rake angle in the EUAT. The thinner chip obtained in EUAT also suggested that the chip deformation was smaller and the shear angle was larger than those in CT, thus leading to a lower cutting force in the EUAT than in the CT.

SUMMARY/CONCLUSIONS

For better understanding the effect of ultrasonic lubrication on the machining characteristics of Ti-6Al-4V by the in-base-plane EUAT method, turning experiments were conducted with/without ultrasonic vibration. The obtained results can be summarized as follows:

- (1) The application of an elliptical ultrasonic vibration in the base plane can significantly lower the cutting force even the cutting speed has exceeded the critical speed limit in traditional elliptical ultrasonic cutting process; the larger the vibration amplitude is, the smaller the cutting force becomes.
- (2) The surface roughness in EUAT increased with the rise of ultrasonic vibration amplitude, which was attributed to the increase of the relative height of the cusp-shaped knurls.
- (3) The ribbon-type chips were formed in CT, while the washer-type chips were formed in EUAT. The thinner chip obtained in EUAT suggested that the chip shear angle was larger than that in CT, thus leading to a lower cutting force than CT.

ACKNOWLEDGEMENTS

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A NOVEL GANTRY 5-AXIS MILLING-TURNING MACHINE TOOL WITH FOUR COLUMNS

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ABSTRACT

A novel gantry 5-axis milling-turning machine tool has moving beam with box-in-box structure was proposed. The symmetrical structure with four columns for improve the stiffness of Z-axis has been designed. In order to meet the milling-turning requirement, a swivel head with direct drive was investigated. The new hydrostatic rotary table has large diameter and thin thickness was developed, and the work area in the vertical direction was raised. The profile error model was established based on homogenous transformation matrix form. The relationship between the moving of axes and the profile errors in the working space was analyzed. The results indicated the y_m is the main factor to cause the change of profile error in working space as the swivel head under the horizontal machining state. The change of A-axis swivel angle has significant influences on profile error.

INTRODUCTION

Due to the characteristics of symmetrical structure, high stiffness and large working space, the gantry-type machine tools have various applications in the machining of accessories and structural components in the aviation and aerospace fields. Bridge type gantry machine tool has the advantages of lighter moving unit and smaller overturning moment during the acceleration process. But the stroke is limited for the reason of the interference between the spindle and the gantry guide rails, and the stiffness of the tool tip declined significantly as the spindle far away the beam. So, bridge type gantry machine tools are primarily suited to the high speed and lightweight processing of aluminium alloy material. The supporting points of gantry machine tools with moving beam located at the bottom of the columns of two sides. So the stiffness will not resulting in a substantial decrease as the spindle nose moving

to the lowest point [1], and no interference occurred within the area of the beam longitude direction. The heavier moving unit will cause lower speed and acceleration of the beam. So, it suitable to the heavy cutting force, like titanium alloy and high-temperature alloys material machining.

In this paper, the gantry 5-axis machine tool MTC2000 is designed for the high precision machining of aviation and aerospace components, which contains milling-turning function. The rotary table diameter is 2000mm. The difficulties of the development process contain the research of 'box-in-box' gantry type structure with four columns, swivel head with horizontal and vertical posture, and the hydrostatic rotary table has large diameter and thin thickness. Also, the profile error in the working space based on homogenous transformation matrix form is investigated.

CONCEPTUAL DESIGN

The box-in-box pattern was employed for the MTC2000, as shown in Figure 1. The material for bed, column and beam is natural granite. The spindle box moved as moving frame driven in vertical direction, which the spindle box can also realize horizontal linear movement. Thermal symmetric structure pattern was employed for the spindle box for reduce the thermal deformation. High precision linear guides were used for X, Y and Z axis. The linear guides of X axis were fixed on bed. Two pairs of linear guides installed symmetrically on the top and two sides of the beam, respectively, for the Y axis. Four columns constituted the symmetric pattern in Z axis, and they each have a linear guide on the outside of the column. The moving beam positioned by the step of Z slider, and connected with Z slider through the end face. Comparing with traditional gantry machine tools,

the stroke of Y axis can be increased in MTC2000. Two oil cylinders were deployed for maintain the equilibrium of the moving beam during the running processor. The torque motor and two reading head were used in the hydrostatic rotary table for improve the positioning accuracy. The height was reduced due to the torque motor was embed into the rotary table. The C-axis and X-axis are structural integration and functional independent. The A-axis is a swivel head which can realize horizontal and vertical machining processes. Also, torque motor and two reading head were used.

A closed 'box-in-box' structure constitute duty columns with lager dimension and the moving beam, which allowed the machine tool to sustain any cutting force in the machining process and prevent the deformation of the structure frame. The tool tip always located in the four columns. One advantage of this pattern is that it can reduce the overturning moment caused by cutting force effectively. In traditional gantry machine tool with two columns, the deformation increased obviously under the same cutting force as the spindle box move upward [1].

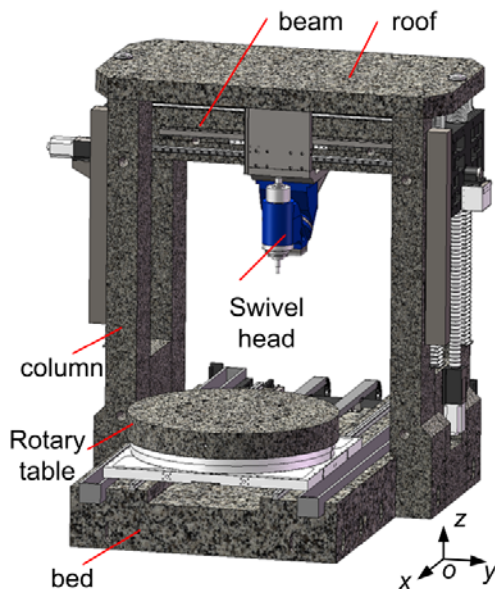


FIGURE 1. Schematic diagram of MTC2000

TABLE 1. Design specifications of the machine tool MTC2000

Parameter	Value
X/Y/Z-axis	2100/2100/1200mm
A-axis Swivel range(0=Vert.)	-90/+90°
C-axis	endless
Table working surface	ø2000mm
Rapid traverse and feed	50 rpm
Rapid traverse and feed	5 rpm
X/Y/Z positioning accuracy	5μm
X/Y/Z re-positioning accuracy	3μm
A/C positioning accuracy	4sec
A/C re-positioning accuracy	2sec
A-axis run-out accuracy	3μm
C-axis run-out accuracy	2μm

DEVELOPMENT

Four columns

Two columns usually used in traditional horizontal or vertical machining centers. The FEM analysis indicated the columns are weak parts, and the deformation of the columns account for about 50% of the total deformation of the gantry machine tool [2]. For improving the stiffness of the machine tool, a four columns structure was proposed, as shown in Figure 2. Both side of the roof has two columns. The parallelism error of the two columns on each side was dependent on the accuracy of adjusting block. The perpendicularity errors between columns and bed were ensured by the form and location tolerance of the locating steps.

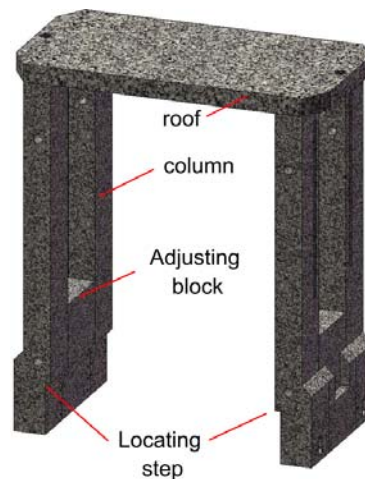
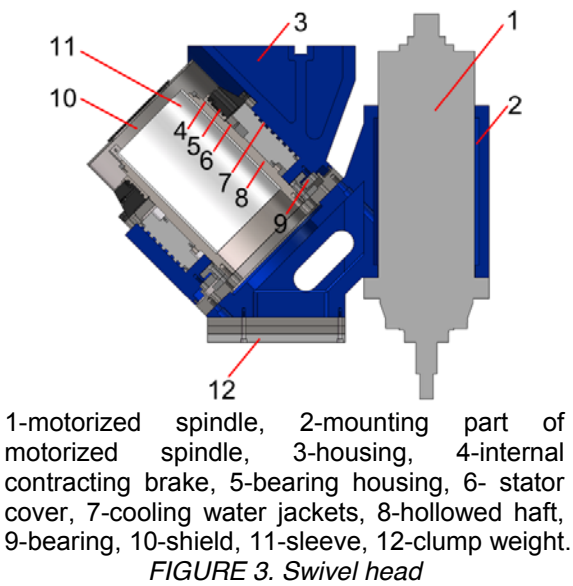


FIGURE 2. Structure of four columns of Z-axis

Swivel head

The key unit in machine tool with horizontal-vertical conversion function is the spindle owns swivel range. Journal bearing and thrust bearing were designed separately and complex in structure in traditional swivel head with horizontal-vertical conversion function. Also, the backlash existed in the speed deceleration mechanism which connected with servo motor and the swivel head will lead to a reduction in the positioning precision.

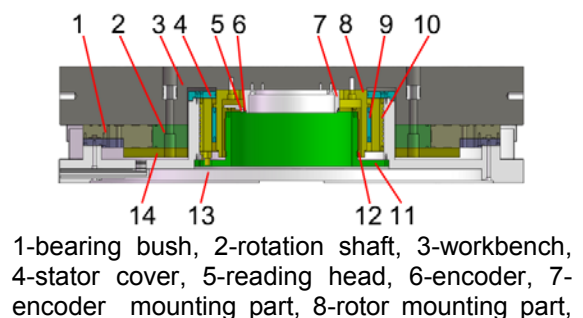
A hollowed swivel head with horizontal-vertical conversion function driven by torque motor was proposed, as shown in Figure 3. The bearing, torque motor, internal contracting brake, and encoder were connected in series along the axis of swivel head. The radial thrust combined rolling bearing was used for supporting the rotating parts. The assemble interface of housing and motorized spindle mounting part is tilted 45 degree relative to the horizontal line. The tool tip lies on the axis of swivel head. Annular pads were machined on the cylindrical surface of the internal contracting brake. Small gap was existed between the inside hole of internal contracting brake and the hollowed shaft. The homogeneous radial deformation occurs as the pressured oil entered into the pads for brake the swivel head. In order to reduce the thermal error induced by torque motor, the cooling water jackets was mounted to an exterior side of stator. The arrangement form contains encoder and two reading head can realize speed control and positioning accuracy improvement.



Rotary table

Due to the characteristics of high precision and high stiffness, the hydrostatic rotary tables have various applications in precision or ultra-precision machine tools [3][4][5]. The hydrostatic bearing and motor were connected in series in traditional hydrostatic rotary table, which induced large thickness and reduced the vertical stroke in machine tools. The journal and thrust bearing were manufactured separately, and the pressured oil was supplied to the respective pads. Besides that, the journal and thrust bearing do not share the same returning oil groove, which adds to the complexity. A hydrostatic rotary table has large diameter and thin thickness was propos in this article, as shown in Figure 4. The torque motor was embed into the inner bore of the hydrostatic bearing which will reduce the thickness of the rotary table. In order to simplify the oil circuit structure, the commonality oil supply pathway and returning oil pathway were designed for hydrostatic journal and thrust bearing. All pads were evenly distributed on the surface of the bearing bush, which enhance the integration of the whole rotary table.

The torque motor stator was fixed at mounting part by flange, and connected to the workbench of rotary table directly. The torque motor stator was designed puts in the cooling water jackets. Axial position of them can be adjusted by changing the axial dimension of stator cover. Seal groove was manufactured on the bearing bush. The internal contracting brake and corresponding mounting part were installed inside the stator mounting part. The homogeneous radial deformation occurs as the pressured oil entered into the pads for brake the rotary table. The encoder and two reading heads were installed in the torque motor where near the workbench of rotary table, which is beneficial to reducing the potential resonance, especially when the servo bandwidth increases.

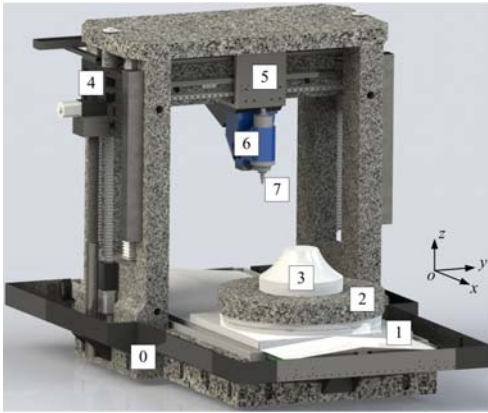


9-torque motor, 10-cooling water jackets, 11-internal contracting brake mounting part, 12-internal contracting brake, 13-base, 14-thrust pad.

FIGURE 4. Hydrostatic rotary table

ANALYSIS AND RESULTS

All relevant error influences are described in the machine error budget. For the present kinematic consideration a homogenous transformation matrix form [3] is developed for the MTC2000. Figure 5 illustrates the topological structure of the machine tool.



0-bed, 1-X-axis, 2-rotary table, 3-workpiece, 4-Z-axis, 5-Y-axis, 6-A-axis (swivel head), 7-tool.

FIGURE 5. Topology structure of MTC2000

The homogeneous coordinate of the tool tip in tool coordinate can be expressed as

$$P_t = (p_{tx} \ p_{ty} \ p_{tz} \ 1)^T \quad (1)$$

And homogeneous coordinate of the workpiece forming point in workpiece coordinate can be expressed as

$$P_w = (p_{wx} \ p_{wy} \ p_{wz} \ 1)^T \quad (2)$$

The tool tip will deviate from the ideal position inevitable due to the various errors of machine tool in actual machining processes. So, the error of the tool tip in the working space can be described as

$$E = \left(\prod_{u=n, L^u(3)=0}^{u=1} T_{L^u(3)L^{u-1}(3)p} T_{L^u(3)L^{u-1}(3)s} \Delta T_{L^u(3)L^{u-1}(3)p} \Delta T_{L^u(3)L^{u-1}(3)s} \right) P_w - \left(\prod_{t=n, L^t(7)=0}^{t=1} T_{L^t(7)L^{t-1}(7)p} T_{L^t(7)L^{t-1}(7)s} \Delta T_{L^t(7)L^{t-1}(7)p} \Delta T_{L^t(7)L^{t-1}(7)s} \right) P_t \quad (3)$$

In Equation (3), the T_{ijp} and T_{ijs} are body ideal static and motion characteristic matrix between body i and j , respectively. The ΔT_{ijp} and ΔT_{ijs} body static and kinematic error characteristic matrix between body i and j , respectively.

Firstly, profile error $f(x_m, y_m, z_m, a_m, c_m)$ in working space under different displacement vectors (x_m, y_m, z_m) of linear axes was analyzed. As $(a_m, c_m) = (\pi/4, 0)$ and $z_m = 0$, the profile error of tool tip in the working space of MTC2000 was shown in Figure 6. It can be concluded that the profile error is sensitive to y_m .

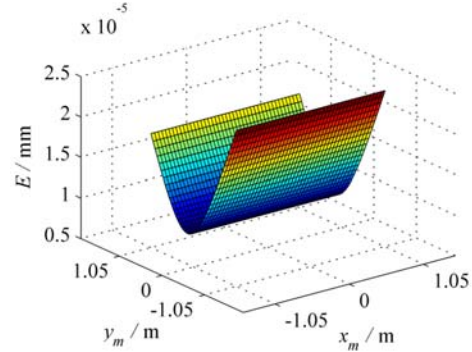


FIGURE 6. Corresponding profile error of $f(x_m, y_m, 0, \pi/2, 0)$

We analyzed the profile error $f(x_m, y_m, z_m, a_m, c_m)$ in working space further considering different angle vectors (a_m, c_m) of rotation axes. As $(x_m, y_m, z_m) = (0, 0, 0)$, the change trends of the profile error in the working space of MTC2000 with various angle vectors (a_m, c_m) were shown in Figure 7. It can be found that the swivel angle of A-axis has significant influence on profile error. The profile errors have minimum value as the MTC 2000 under the horizontal and vertical machining states, it also means the swivel angle of A-axis are 0 and 90 degree. As the swivel angle of A-axis reach 45 degree, the maximum profile error occurred. Profile error is showing a tendency to increase progressively in 0-45 degree range, and reduce progressively in 45-90 degree range. The trend of the profile error curve shows symmetrical distribution in the swivel range of A-axis.

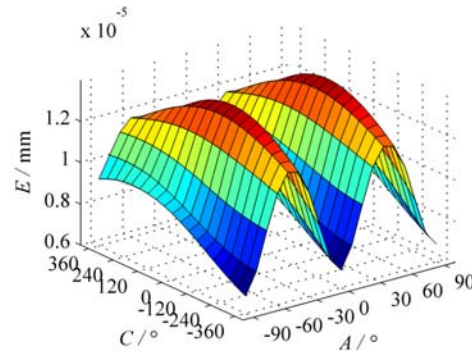


FIGURE 7. Corresponding profile error of $f(0,0,0,a_m,c_m)$

As $(x_m, y_m, z_m) = (1.05, 1.05, 0.6)$, which the displacement of linear axes reach the maximum value, the change trends of the profile error in the working space of MTC2000 with various angle vectors (a_m, c_m) were shown in Figure 8. The profile error increase significantly compare with situation of $(x_m, y_m, z_m) = (0, 0, 0)$.

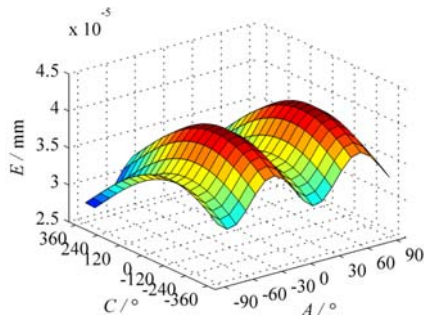


FIGURE 8. Corresponding profile error of $f(1.05, 1.05, 0.6, a_m, c_m)$

CONCLUSIONS

A novel gantry machine tool structure was proposed in this paper. The machine and key units were developed. The box-in-box pattern with moving beam was employed for MTC2000. A swivel head with direct drive was investigated and a new hydrostatic rotary table has large diameter and thin thickness was developed. A four columns structure for improve the stiffness of the machine tool was proposed.

As the rotation axes have explicit displacement vectors, the y_m is the main factor to cause the change of profile error in working space as the swivel head under the horizontal machining state. The A-axis swivel angle has significant influences on profile error when the linear axes have explicit displacement vectors.

ACKNOWLEDGEMENT

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TOOL FLANK WEAR EFFECTS ON THE CUTTING EDGE SHRINK IN ULTRA-PRECISION RASTER FLY CUTTING

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INSTRUCTIONS

In metal cutting, tool wear is an unavoidable phenomenon. There are many factors affecting the occurrence and characteristics of tool wear such as tool materials, cutting condition, tool geometry, and the workpiece materials being cut. In conventional cutting, the tool wear characteristics include: Nose wear, tool face wear, wear-land wear, cratering, and plastic flow [1]. However, the tool wear characteristics in ultra-precision machining are different from conventional machining. These differences include two aspects, first the cutting tool in ultra-precision machining is single crystal diamond, who owns super hardness and high thermal conductivity, which make the tool wear characteristic different [2]. Second, the diamond turnable workpiece material is broad, it not only include ductile materials like aluminum, copper, electroless nickel, but also include some brittle materials like ceramics, silicon, glass and so on [3-7]. Similar to conventional cutting, in ultra-precision cutting, tool wear occurs and features largely depend on the cutting condition, the workpiece material being cut. Cutting condition can seriously affect the tool life and tool wear characteristics. However, for diamond cutting ductile material such as copper and aluminum, a smooth and flat wear land is formed [8].

Ultra-precision raster fly cutting (UPRFC) is a typical intermittent cutting process whereby the diamond tool rotates across the spindle while the workpiece is fixed. The intermittent cutting property of UPRFC makes its tool wear characteristics different from other cutting process. This difference includes two aspects: one is that diamond tools are employed in the UPRFC cutting process, the extreme hardness and low friction coefficient of diamond tools delays the occurrence of tool wear [9]. The other

one is that the discontinuous cutting mechanism of UPRFC reduces contact time between the diamond tool and workpiece, which allows for superior heat dispersion to suppress the occurrence of tool wear [10]. Until now, Yin et al. (2009) has conducted the research on tool wear characteristics in UPRFC, they identified the characteristics of diamond tool as fractures or micro chipping due to effects of impacts during the UPRFC process [11]. Our previous study found that the tool wear characteristics of UPRFC not only includes fractures of the cutting edge at the initial tool wear stage, but also includes material welding on the rake face, wear land and sub-wear-land formation at the steady wear stage, and micro-groove on the clearance of cutting tool [12]. Moreover, the tool wear characteristics identification methods and tool wear effects on the machined surface quality are investigated [13-15]. However, studies of tool flank wear effects on the cutting edge receding in UPRFC have not been conducted before.

In the present research, the tool flank wear characteristics are investigated, the effects of tool flank wear on cutting edge shrink is modeled, and cutting edge shrink effects on the surface roughness are discussed. The present research provides a deep insight into the relation between tool flank wear and cutting edge in ultra-precision machining.

EXPERIMENTS

In this study, a Precitech Freeform 705G (Precitech Inc., USA) five-axes CNC ultra-precision lath was used to perform the cutting experiment, which owns two typical cutting strategies: horizontal cutting and vertical cutting, as is shown in Fig. 1. In this study, a diamond tool designed and fabricated by Apex Inc., UK was chosen to cut the desired flat surface. Tool

geometry parameters are: tool nose radius of 0.631mm, rake angle of -2.5° , clearance angle of 15° . Cutting parameters used in this study are: feed rate: 200mm/min, depth of cut: 0.03mm, spindle speed: 4500rpm, swing distance:

28.35mm, step distance: 0.025mm. Cutting environment is lubricant on, cutting strategy is horizontal cutting. The workpiece material used in this experiment is brass. The total cutting distance is around 5000 meters.

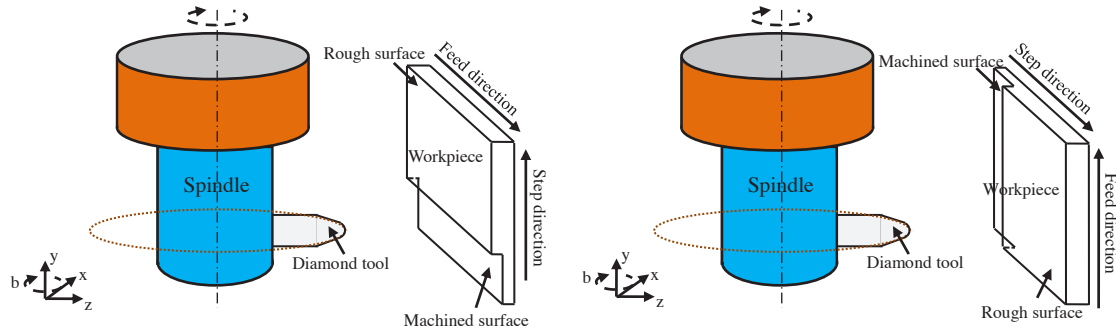


FIGURE 1. Cutting strategies in UPRFC

After cutting, the diamond tool was dismantled and inspected by Hitachi TM3000 scanning electron microscope (SEM), while the cutting edge profile was measured by a Park's XE-70 atomic force microscope (AFM).

RESULTS AND DISCUSSION

Tool Flank Wear Features

Diamond material has excellent mechanical properties such as extreme hardness and low friction coefficient. However, with the growing of the cutting distance, the mechanical wear of diamond tool is evitable. Fig.2 shows SEM tool wear figures after 5000m diamond flat cutting, it is found a smooth wear land is formed on the cutting edge. The wear land is wider at its central while thinner at its two sides, which is figured as crescent-like profile.

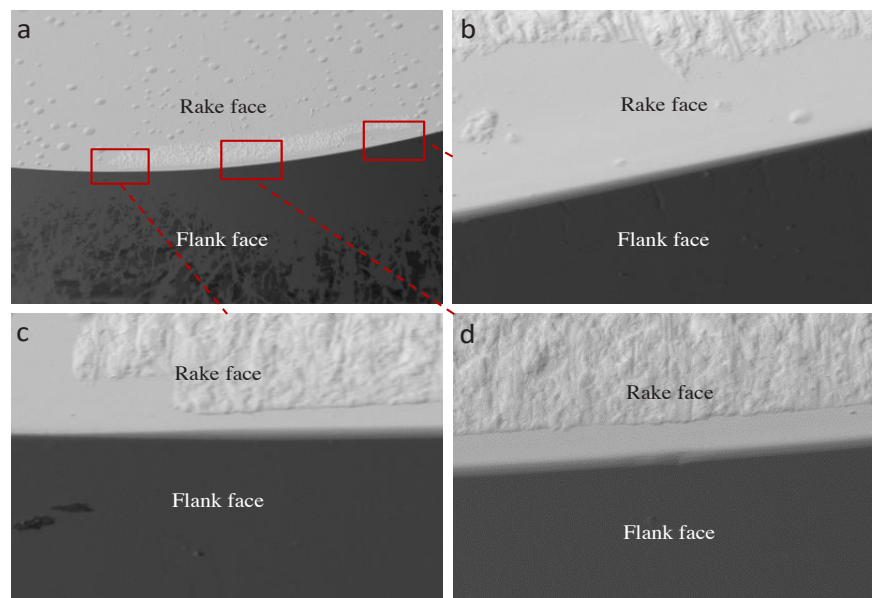


FIGURE 2. SEM figures of tool flank wear patterns after 5000m flat cutting

The occurrence of tool flank wear makes the cutting edge shrink and changes the tool geometry parameter. Therefore it will affect the form accuracy of machined surface.

Fig.3 shows the AFM figure of cutting edge, it is found that after a long distance cutting, a smooth wear land is formed on the cutting edge. The wear land has an angle ϕ with the rake face of diamond tool.

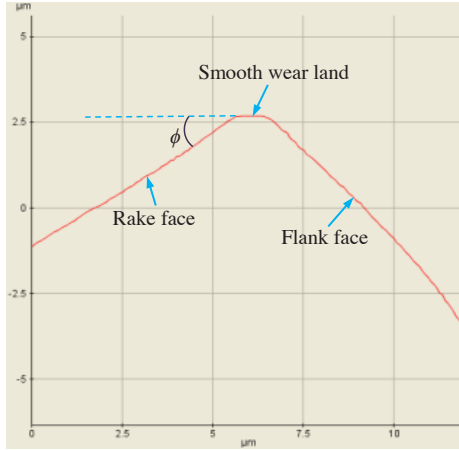


FIGURE 3. AFM photograph of cutting edge

Based on Fig.3, the width of wear land can be measured. Also the wear land angle can be measured.

Cutting Edge Shrinks Model

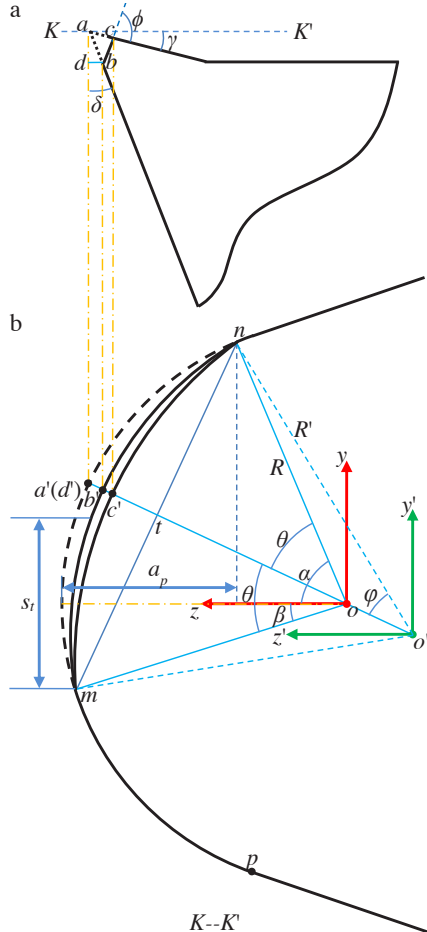


FIGURE 4. Schematic of the side view (a) and the top view(K--K' plane) (b) of tool flank wear

For clarity, two coordinate systems $o-yz$ and $o'-y'z'$ were established to present different arcs, as shown in Fig. 4 (b). In Fig. 4 (a), $|bc|$ is the wear land width, whose value can be identified from the cutting chips, and ϕ is the wear land angle, whose value is 40.5° since brass material being cut.

Based on the geometric relationship, the cutting edge retraction ($|bd|$) can be calculated from the Sine law. As is shown in Fig.4 (a), in triangle $\triangle abc$, $\angle acb = \phi$, $\angle bac = 90^\circ - \delta - \gamma$, according to the Sine law:

$$\frac{|bc|}{\sin(90^\circ - \delta - \gamma)} = \frac{|ab|}{\sin \phi} \quad (1)$$

From Eq. (1), the length $|ab|$ can be calculated as:

$$|ab| = \frac{\sin \phi}{\sin(90^\circ - \delta - \gamma)} |bc| = \frac{\sin \phi}{\sin(90^\circ - \delta - \gamma)} w \quad (2)$$

Where w is the wear land width.

In triangle $\triangle abd$, the length $|bd|$ can be calculated as:

$$|bd| = |ab| \sin \delta = \frac{\sin \phi \sin \delta}{\sin(90^\circ - \delta - \gamma)} w \quad (3)$$

In the thrust plane $K-K'$, supposing $h_t = |b'd'|$, we obtain:

$$h_t = |b'd'| = |bd| = \frac{\sin \phi \sin \delta}{\sin(90^\circ - \delta - \gamma)} w \quad (4)$$

New Formed Tool Nose Radius

The tool flank wear in the UPRFC process makes the tool material loss-zone a crescent-like profile that is thicker at the center position while thinner at the two sides, as shown in Fig. 4 (b). Suppose the newly formed cutting edge and the original cutting edge are intersected at the point m and n , while the central point of new cutting edge arc mn is b' .

In Fig. 4 (b), according to the geometric relationship, we obtain:

$$\begin{cases} \cos \alpha = (R - a_p) / R \\ \sin \beta = \frac{s_t}{2R} \end{cases} \quad (5)$$

From Eq. (5), two angle α and β can be calculated as:

$$\begin{cases} \alpha = \arccos\left(\frac{R - a_p}{R}\right) \\ \beta = \arcsin\left(\frac{s_t}{2R}\right) \end{cases} \quad (6)$$

Therefore, θ in Fig. 4 (b) can be calculated as:

$$\theta = \frac{\alpha + \beta}{2} \quad (7)$$

In triangle ont in Fig. 4 (b), $|nt|$ can be solved as:

$$|nt| = R \sin \theta \quad (8)$$

Similarly, in triangle o'nt, $\sin \varphi$ is calculated as:

$$\sin \varphi = \frac{|nt|}{R'} = \frac{R \sin \theta}{R'} \quad (9)$$

From Eq. (9), the angle of φ can be computed as:

$$\varphi = \arcsin\left(\frac{R \sin \theta}{R'}\right) \quad (10)$$

Since the difference of $|a't|$ and $|b't|$ equals the height of the loss zone (h_l), yields:

$$(R - R \cos \theta) - (R' - R' \cos \varphi) = h_l \quad (11)$$

Submitting Eq. (10) into φ in Eq. (11) yields:

$$R(1 - \cos \theta) - R' \left[1 - \cos \left(\arcsin \left(\frac{R \sin \theta}{R'} \right) \right) \right] = h_l \quad (12)$$

Eq. (12) can also be written as a function, as:

$$F(R') = R(1 - \cos \theta) - R' \left[1 - \cos \left(\arcsin \left(\frac{R \sin \theta}{R'} \right) \right) \right] - h_l = 0 \quad (13)$$

From Eq. (13), R' can be solved from numerical methods.

Effect Of Cutting Edge Shrink On The Machined Surface Roughness

Fig. 5 (a) shows that the surface topography is formed by the imprints of the cutting edge and that in the topography forming process, the intersection point of two original cutting edges is point g . However, the retraction of the cutting edge due to tool flank wear leads to the new intersection between the original cutting edge and worn cutting edge; the intersection point has been changed into point u , which causes a motion of the intersection point. In actuality, the retraction of cutting edge and the motion of intersection point are progressing simultaneously. Therefore, the retraction of cutting edge leads to the intersection of worn cutting edge at point u . Due to the bigger cutting edge radius of the worn cutting edge, the imprints of the worn cutting edge on the machined surface will cause a smaller surface roughness, as shown in Fig. 5 (b).

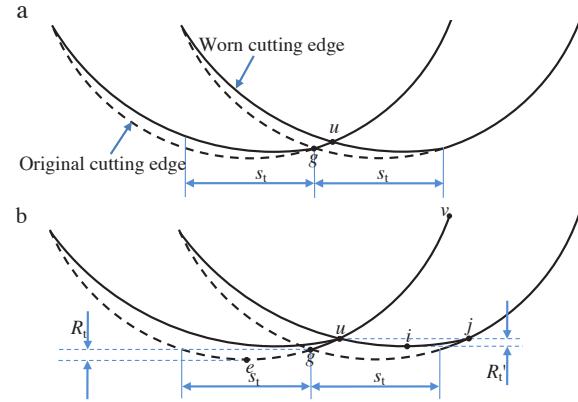


FIGURE 5. Tool flank wear effect on surface roughness

The theoretical peak-to-valley roughness in UPRFC is calculated as [14]:

$$\begin{cases} R_{t_min} = R_{t_tool} = R - \sqrt{R^2 - (s_t / 2)^2} \\ R_{t_max} = R_{t_swing} + R_{t_tool} / \cos \alpha = \\ s_w - \sqrt{s_w^2 - (f_e / 2)^2} + \frac{s_w (R - \sqrt{R^2 - (s_t / 2)^2})}{\sqrt{s_w^2 - (f_e / 2)^2}} \end{cases} \quad (14)$$

As the tool flank wear occurs, a new tool nose radius (R') is formed, the theoretical peak-to-valley roughness is changed into:

$$\left\{ \begin{array}{l} R'_{t_min} = R' - \sqrt{R'^2 - (s_t / 2)^2} \\ R'_{t_max} = s_w - \sqrt{s_w^2 - (f_e / 2)^2} \\ + \frac{s_w \left(R' - \sqrt{R'^2 - (s_t / 2)^2} \right)}{\sqrt{s_w^2 - (f_e / 2)^2}} \end{array} \right. \quad (15)$$

CONCLUSIONS

In the present research, cutting edge shrink under tool flank wear and its effects on the machined surface roughness are investigated. Some specific conclusions drawn from the research are:

1. A smooth wear land is formed on the cutting edge as diamond cutting ductile materials, the smooth wear land has an angle with the rake face of diamond tools.
2. The occurrence of tool flank wear forms a new tool nose arc with a larger radius and a changed circle center.
3. The formation of tool flank wear affect the machined surface roughness, theoretically, cutting edge shrink reduce the surface roughness.

ACKNOWLEDGEMENTS

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RESEARCH ON THE ATMOSPHERIC PRESSURE PLASMA PROCESSING OF MICRO FREEFORM OPTICAL SURFACE ON FUSED SILICA SUBSTRATE

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INTRODUCTION

Fused silica is widely used in the optics system because of its excellent optical properties and thermal stability. For example, in the final optics system of inertial confinement fusion (ICF) in National Ignition Facility Project (NIF), various types of optics are made of fused silica, such as the wedged focus lens of non-uniform thickness provided color separation via lateral dispersion and the continuous-contour phase plate of high spatial resolution for smoothing the focal spot profile [1]. How to fabricate low damaged or zero-defect optical surface becomes a challenge nowadays.

The atmospheric pressure plasma processing (APPP) is a technology developed from the last decade in 20th century for non-mechanical ultra-precision surface shape generation with low surface damaged layer, which is easily applied to generate optical surfaces but also to improve the surface shape accuracy and reduce the surface damaged layer in a very fast and cost-efficient way [2]. Also, this technology is named different appellation by some research institutes all over the world respectively. For example, reactive atom plasma (RAP) technology was presented by Lawrence Livermore National Laboratory (LLNL) of US, then, RAPT industries Inc. has developed the apparatus and method for shaping damage-free surfaces such as fused silica, single crystal silicon and silicon carbide with the help of inductively coupled plasma (ICP) at atmospheric pressure generated by a radio frequency power source with the addition of fluorocarbons or other types of reaction gases [3]. Meanwhile, the Chemical Vapor Machining (CVM) process has been developed by Osaka University of Japan, in which local plasmas are generated in the vicinity of rotary or pipe electrodes powered with high frequency at 150 MHz [4]. On the other side, the Atmospheric Plasma Jet Machining (PJM) technology was also proposed by the Leibniz-Institute of Surface

Modification in Germany utilizing a microwave power source and capacitively coupled plasma (CCP) to accomplish similar processing [5]. The purpose of all processes mentioned above is to fabricate the deterministic surface shape of optical parts with high form accuracy and low surface damaged layer.

In this paper, a five-axis coordinated numerical control system for APPP was developed; meanwhile a needle electrode torch and a microelectrode torch for plasma generation were designed. Then, the APPP method with CCP is adopted to realize the fabrication of micro freeform optical surface on fused silica substrate, which could provide stable removal function, high material removal rate and processing resolution without surface damage. So, it makes the APPP method with a great potential to fabricate micro freeform optical surface on fused silica substrate.

REACTION MECHANISM OF APPP

The reaction and removal mechanism of a silicon based material like Si, SiO₂, or SiC on the substrate, is realized by chemical reactions of atomic fluorine with silicon, or by the reaction of fluorine and oxygen with silicon carbide forming the volatile reaction products SiF₄ and CO₂, respectively. APPP is based on the reaction of a non-thermal, chemically reactive plasma jet driven either with microwave or RF power under atmospheric pressure conditions. The plasma jet is fed by a mixture of fluorine containing reactive gas (NF₃, CF₄, and SF₆) and oxygen. The reactive gas dissociates in the plasma producing chemically reactive fluorine radicals and other highly reactive species that react with the workpiece surface to form volatile compounds, which are exhausted [5]. In the case of fused silica substrates, SiF₄ and CO₂ are formed as a waste product. The reaction mechanism of APPP is as shown in Fig.1.

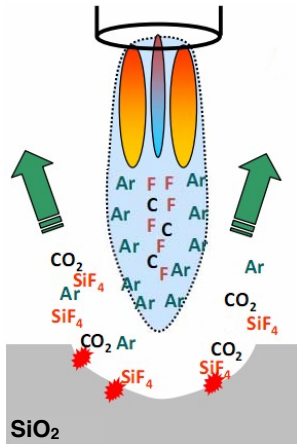


FIGURE 1. Reaction mechanism of APPP

In the experiment, a local white phenomenon at etching points was generated on the fused silica surface, which is changeable with variation of different processing value. When the flow rate of He is 674sccm and 2021sccm, the form of the etching points were distinguishing as shown in Fig.2.

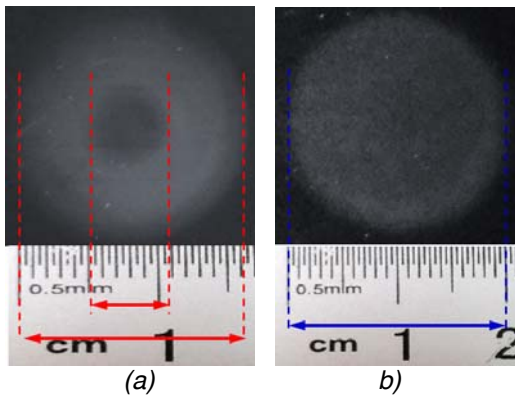


FIGURE 2. Etching points with different flow rates of He: (a) flow rate of He is 674sccm; (b) flow rate of He is 2021sccm.

In order to explain the reason why the local white phenomenon generated on the fused silica surface after APPP machining, the chemical composition, surface morphology, cross-section micro-profile and surface roughness of the processing areas are detected and analyzed respectively. It is proved that the white opaque area is caused by the overlarge surface roughness, not caused by the deposition of intermediate products of chemical reactions, as widely believed. The distributions of excited atoms on the surface are solved. The formation mechanism of surface morphology is verified on the aspect of surface chemical reaction.

EXPERIMENTAL SETUP

A five-axis coordinated numerical control system for APPP was developed, which includes the machine, the RF power supply and matcher, gas supply and control apparatus, tail gas treatment system, motion controller, plasma torch and in situ monitoring devices. There are three translational motion axes, X-axis, Y-axis and Z-axis, with two rotational motion axes, A-axis rotation around X direction and C-axis rotation around Z direction in the machine as shown in Fig.3 (a). Normal, the optical workpiece is fixed on the horizontal X-Y workbench, and the plasma torch is fixed on the front end of Z-axis, which could move horizontally in Z direction and rotate in both A and C direction. The reaction of fluorine radicals with fused silica substrate during the whole process was in a closed chamber to avoid the exhausted gas leakage.

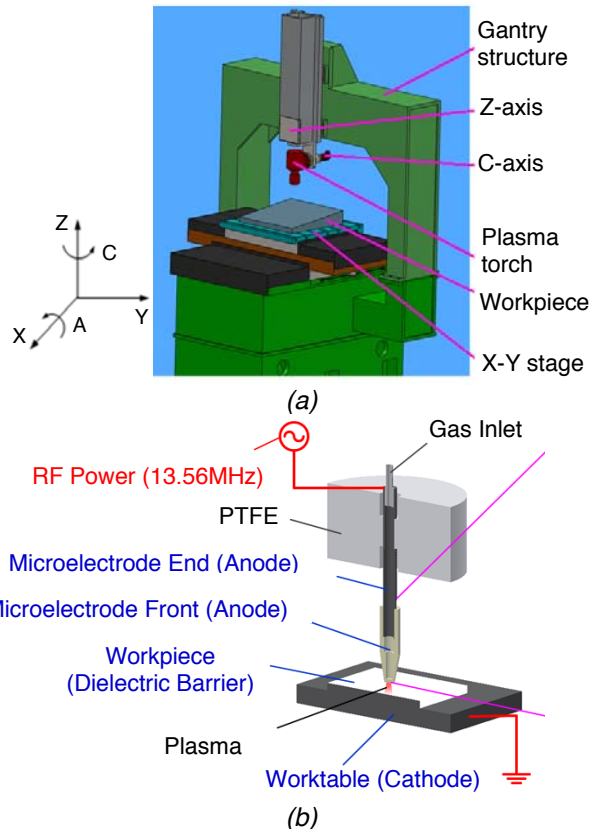


FIGURE 3. Numerical control machining system of APPP: (a) 5-axis coordinated machine for APPP; (b) structure of microelectrode torch.

According to the principle of Dielectric Barrier Discharge (DBD), two kinds of plasma torch were designed and produced, which is needle electrode torch and microelectrode torch. As

shown in Fig.3 (b), a radio frequency (RF) power at 13.56 MHz was used on the microelectrode torch as the anode, worktable was the cathode and the workpiece between the microelectrode and worktable was the dielectric barrier layer. For the microelectrode torch with diameter of 0.2mm, the experimental parameters of APPP were shown in Tab.1.

TABLE 1. Processing parameters of $\Phi 0.2\text{mm}$ microelectrode torch.

Parameter	Value
Rate of He flow (sccm)	810
Rate of CF_4 flow (sccm)	48
Distance (mm)	1
Power (W)	101-103
Time (min)	2

When the plasma eject with the gas flow from the front of the microelectrode, we can see from Fig. 4 that the removal profiles were of a Gaussian shape. The removal rate in depth and full width at half maximum (FWHM) were 590nm/min and 0.3mm.

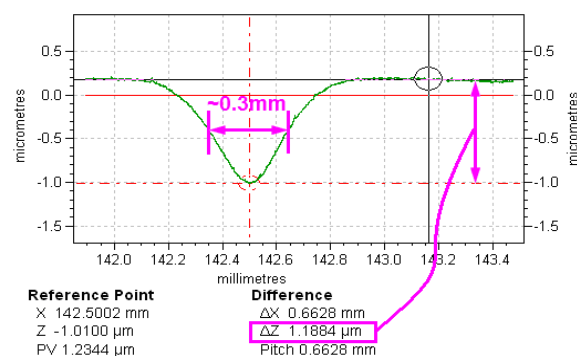


FIGURE 4. Typical removal profile of $\Phi 0.2\text{mm}$ microelectrode torch.

FABRICATION OF MICRO FREEFORM OPTICAL SURFACE

Considered the material removal rate of fused silica substrate mentioned above, meanwhile taking both the processing efficiency and surface accuracy into account, the multi-aperture combination processing method is proposed to fabricate the micro freeform optical surface. Firstly, the surface figure is analyzed to match with the removal functions of plasma torches. Secondly, the mathematical model of multi-aperture combination processing is established. Optimization solutions of multiple removal functions are accomplished during one

iteration process to obtain both high efficiency and accuracy. Thirdly, the solving algorithm is designed to get the dwell time matrixes for plasma torches. The algorithm has a fast convergence speed, which can ensure the processing accuracy and get high computation efficiency at the same time. Finally, the deterministic processing of surface figure is realized by controlling the dwell time of plasma torches on workpiece surface.

In order to ensure the surface figure accuracy, the error sources of APPP are analyzed, and the effective processing methods for improving the surface figure accuracy are presented, which can reduce the machining error caused by the change of temperature. When the processing parameters are shown in Tab.1, the sinusoidal grid with period of 2mm & amplitude of 100nm, one kind of micro freeform optical surface, was fabricated on fused silica substrate with size of 6mm $\times$ 6mm in 64.3 minutes, shown in Fig.5.

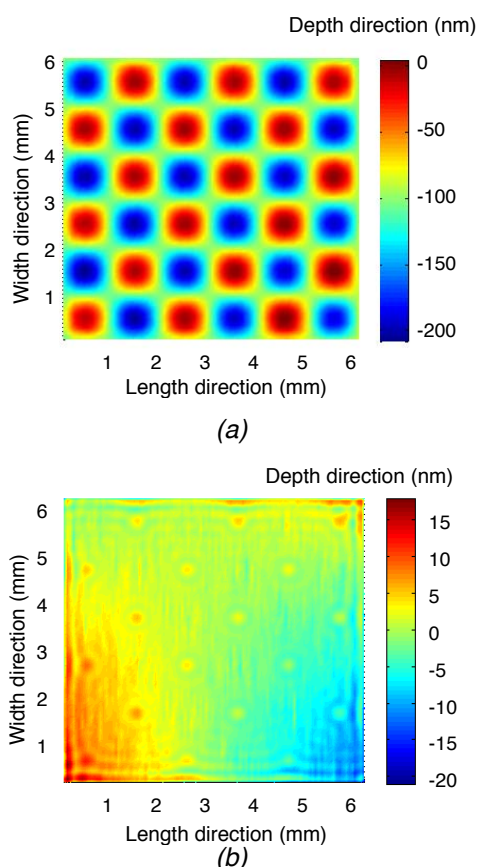


FIGURE 5. Fabrication of sinusoidal grid with period of 2mm & amplitude of 100nm after error improvement: (a) processing result after error improvement; (b) processing error.

The result of processing error shown in Fig.5 (b) was analyzed; it was found that the peak-to-valley value of error was 22.3nm and the root-mean-square value of error was 3.2nm.

Then, the continuous-contour phase plate (CPP), as shown in Fig.6 (a), another kind of micro freeform optical surface, was fabricated with parameters mentioned above with the size of 36mm×36mm in 37.4 minutes.

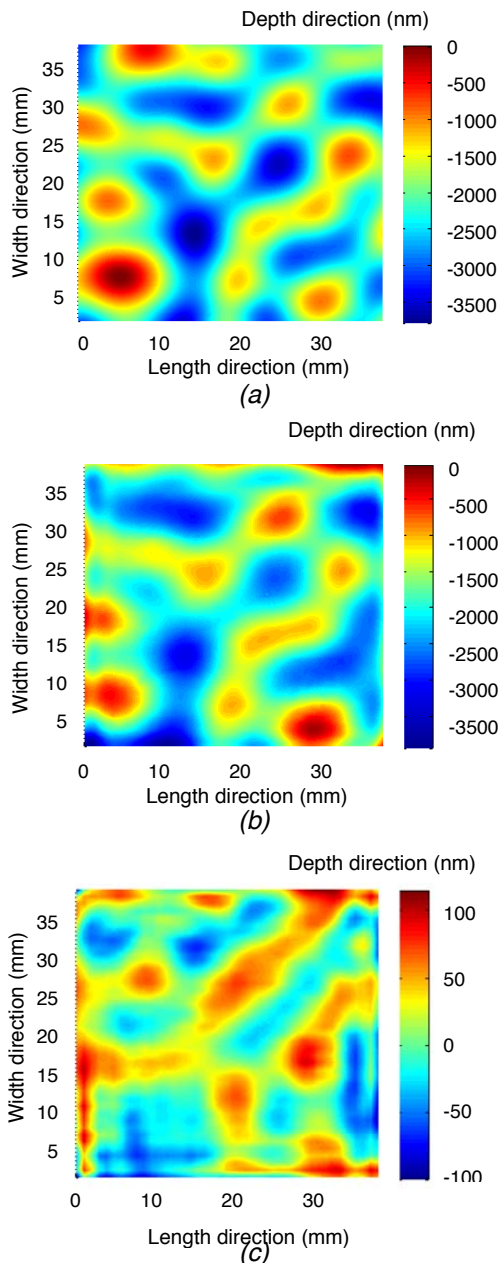


FIGURE 6. Fabrication of micro freeform optical surface: (a) objective surface figure; (b) surface figure obtained from APPP; (c) processing error.

The surface figure and processing error were also obtained as shown in Fig.6 (b) and (c), from which it was found that the peak-to-valley value of error was 163.4nm in the size of 30mm×30mm, and the root-mean-square value of error was 31.7nm.

SUMMARY

Reaction mechanism of atmospheric pressure plasma processing is introduced briefly. Then, a numeral control machining system of APPP and two types of plasma torch are developed. In order to ensure the surface figure accuracy of micro freeform optical surface, the error sources of APPP are analyzed, and the effective processing methods for improving the surface figure accuracy are presented. At last, two kinds of micro freeform optical surface, the sinusoidal grid and the continuous-contour phase plate, are fabricated by the APPP technology of which shows a great potential to fabricate micro freeform optical surface on fused silica substrate.

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Micro V Groove Ruling on Convex Surface by Diamond Tool

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ABSTRACT

This paper explains and demonstrates how to machining micro V groove on convex surface in diamond tool lathe by ruling, it includes optical parts preparation, tool calibration, ruling path program, NC coding, V groove ruling and measurement. The technologies can be the reference for similar application.

Keywords: V groove, diamond tool, ruling, cutting process

1. INTRODUCTION

A part is required as fig.1. its material is PMMA, diameter is 20 mm. The important machining processes are including: first is tool calibration. we used one tool cutting both the convex curve and the V groove. Second is spindle balance. Third is NC program preparation. They are the convex cutting G code and V groove ruling program. Fourth is cutting the convex curve first, then cutting the V groove in C axis mode.

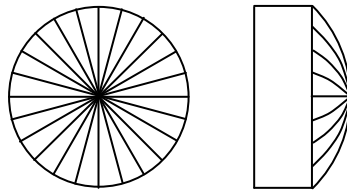


Fig.1 Diagram of the part

2. TOOL CALIBRATION

Mounting a diamond tool on the tool post, the tool is made by Contour Fine Tooling Ltd and its nose radius is 0.036. The optical tool set (OTS) provided by the diamond turning can be used to align the geometric center of the tools to the axis of the spindle shaft. The OTS consists of CCD camera, frame grabber, wiring and control software, etc. the tool adjustment procedure is that: mount the tool on the tool post, make the lens of the camera focus on the tool, that is shown in Fig.2. The image of the tool can be seen on the screen, it is shown in Fig.3. Move the cross cursor along the circular arc of the tool and click at least three points. The system can calculate the center point of circular arc, the X, Y, Z coordinates, which are the distance from the machine X, Y and Z axis home positions to the OTS center. After the adjustment, it is also the distance from the machine home to the tool nose arc center.



Fig.2 Optical Tool Set (OTS)

Fig.3 Tool Nose Image

Unload the data measured by OTS into the tool system T0101, when the NC program is processed, the data will be activated. The data measured by OTS are only the preliminary X, Y, and Z coordinates parameters, in actual application, it needs to machining a workpiece sample, and measure the actual position of the tool. During the repeat sample measuring, X and Y axis coordinates adjusting and sample machining, finally the tool position can be settled. The purpose of this paper is processing the V groove by ruling, whether the tool nose is exactly in the spindle center or not is not important, so the tool position settled by OTS is acceptable.

Workpiece fixture is gently loaded onto the spindle vacuum chuck. Workpiece is a cylinder, mounted on fixture by screw as shown in Fig.4. the material is aluminum 6061. After mounting on spindle, it must center align with the spindle by click the key GORGE, and used the index, to gently adjust the position of the fixture until the workpiece centering with the spindle shaft.



Fig.4 Part is fixed in spindle

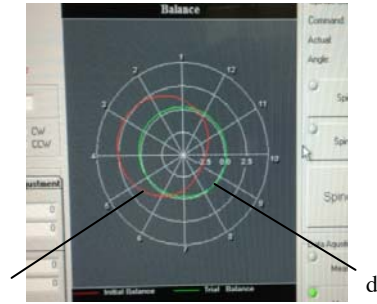


Fig.5 Spindle Balance

As the workpiece or fixture is relatively weight, it must be in dynamic balance. The procedure is that: select BALANCE in the panel of the operator system. Set the spindle speed, normally the face cutting speed is 2000rpm clockwise, so the speed of dynamic balance is also chosen as 2000rpm clockwise. Note the start angle of the spindle. Turn on the spindle and check the error of balance. If the spindle balance is unsatisfactory, adjust the weight screw, turn on the spindle again, and check the balance result. Repeat the procedure several times, until the spindle is balanced. It is shown in Fig.5.

Set the X, Y and Z axis back to machine home position.

3. CONVEX SURFACE MACHINING

The segment a-c-o represent workblank, the final part is curve a-o' as shown in Fig.6, the trajectory created by NanoCAD is b-c-d-h, what the tool travels along the trajectory and feed a depth of cut in z axis is one loop, because the amount of feed is very small, it is approximately 5 micro, it needs more than several hundred loops all together, so it will waste a lot of time when machining in the area d-h-g-f'-e'-e-d'-d.

A feasible method to shorten the time consumption is to cut the only inside zone of b-c-d-d'-e-e'-f-f'-g-o'-a. It needs to set several zone flag, such as d-d', e-d', f-f'. when faced with the zone flag during cutting along the trajectory as shown in Fig.7, the loop will be stopped. After feeding a depth of cut, it starts another loop automatically.

3.1 Zone flag setting

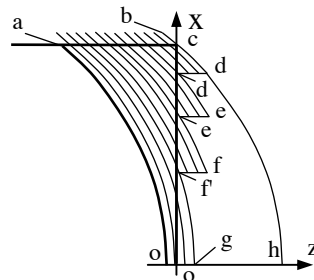


Fig.6 Convex trajectory

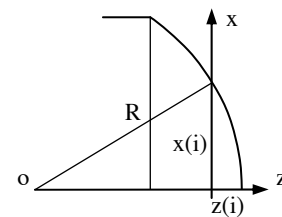


Fig.7 X coordinate of zone flag

The amount of loop can be divided into several parts, the more the amount of loop be divided, the less cutting time consumption, and also the more human operation be needed for the NC program treatment. For appropriate consideration, loop is divided into four parts.

The X coordinate of zone flag (d'-d, e'-e and f'-f) must find. Take a section circular arc as example, it can be seen in fig.3.

$$x(i)=[2R \times z(i) - z(i)^2]-1/2 \quad (1)$$

$z(i)$ is one third of the amount of loop, or one fourth of the amount of loop multiply with the feed of cut in z axis.

$$z(i)=\text{amount of loop} \times \text{feed of cut} \times i \quad (2)$$

$$i=1/4, 1/2, 3/4$$

If the R is 4.4225, the amount of loop is 400, the feed of cut is 5 micro, the $z(i)$ can be calculated from the formula (2), and take the $z(i)$ into the formula (1), $x(i)$ can be obtained. it is shown in table 1.

Table 1 x coordinate of Zone flag

Flag, i	Flag1, 1/4	Flag2, 1/2	Flag3, 3/4
$z(i)$	0.5	1	1.5
$x(i)$	2.043	2.801	3.319

The actual value should be chosen a little larger than that listed in table 1.

3.2 Loop jump

Conditional block execution allows the programmer to execute a statement based on a conditional expression. This is accomplished with the IF statement. It can be used as following:

IF [<#integer1>LT<integer2>]GOTO<integer3>

Integer is a positive integral number, integer1 is the loop count.

Integer2 is a part of the loop count, for example they can be 1/4, 1/2, 3/4 loop count.

Integer3 is the position in the end of the program. if the conditional operator is true, the program execution will jump to the intrger2 position.

In this paper, the number of loops is 400 (i.e. #501= 400), the loop of count is set to zero initially (i.e. #506=0), only when the loop of count has been added to 400, the cycle of the program stops. The number of loops is divided into 4 parts, so the integer2 are 100, 200, 300. the integer3 is 2, so the IF statement are:

IF [#506LT100]GOTO 2

IF [#506LT200]GOTO 2

IF [#506LT300]GOTO 2

Where to embed the IF statements into the NC program depends on the X coordinate of Zone flag, they are in X2.043, X2.801, X3.319.

What the flow chart appears in the NC program can be seen as below.

```
#501= 400      ( NUMBER OF LOOPS )
#506 = 0      ( LOOP COUNT )
N3
.....
X2.043 Z..... (zone flag1)
IF[#506LT100]GOTO 2
.....
X2.801 Z..... (zone flag2)
IF [#506LT200]GOTO 2
.....
X3.319 Z..... (zone flag3)
IF [#506LT300]GOTO 2
```

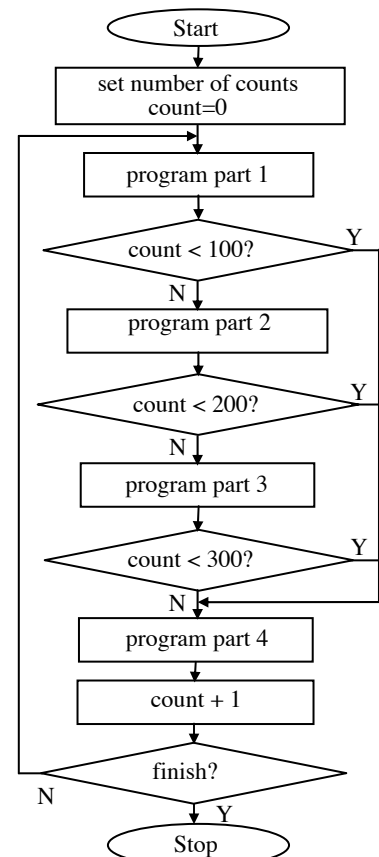


Fig.8. The flow chart of the NC program for loop jump

The flow chart can be seen in Fig.8.

Use the same tool to ruling the convex surface along the Z-Y plane, so the tool moves along Z and Y axis coordinately. The V groove will be machined in vertical direction. The ruling path is as same as the path of convex surface machining, but all Y axis data is replaced by X axis data, NC program is almost the same program with the convex surface machining, the main difference is deleting the line:

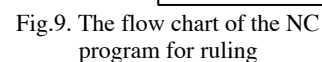
And adding the following lines:

G09 C0.0 (USER INPUT, C AXIS POSITION)

Initialing: set all cut variables, such as number of loops, depth of cut, loop count, etc.

each groove can be deeper as well by ruling several times, each time the Z axis get a depth of cut.

The feed rate is 6mm/min. The coolant is mist mixed inorganic alcohol with gas.



Micro V groove can be machined by ruling used Moore Nanotech 350 FG diamond turning, there are several procedures must go through that spindle balance, tool centering adjustment, NC programming, convex surface cutting, and V groove ruling. The finished part is shown in fig. 10.

MEASUREMENT NOISE OF A COHERENCE SCANNING INTERFEROMETER IN AN INDUSTRIAL ENVIRONMENT

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INTRODUCTION

Coherence scanning interferometry (CSI) has seen a rapid uptake in manufacturing industry for use in quality control of parts that require surface texture to be controlled to the micrometre and nanometre levels. This uptake is largely due to the relative speed of measurement of CSI compared to other techniques, as well as its claimed potential for nanometric height accuracy [1].

In some applications of CSI, the user's understanding of the operating principles of the instrument can be low – technicians who are not specialised metrologists will often use the instrument as a “black box”. As with all metrology instruments, CSIs can be sensitive to changes in measurement conditions. With the increase in interest in surfaces controlled to nanometric tolerances, the potential for unknown effects is likely to become an increasingly serious issue for the CSI users.

Efforts have been made by various national measurement institutes (NMIs) and standardisation bodies to provide guides and standard operating procedures for CSI users (see for example, [2] [3]). The intention of these guides is to educate non-specialists in some of the aspects of the instrument and its use that need to be considered, such as the measurement noise.

The fact that the work towards guides commonly takes place within the NMIs, means that the guides are largely compiled using instruments in very stringent conditions; frequently housed away from as many sources of noise as possible. This is done so as to provide the most accurate measurements possible and enables NMIs to identify only issues implicit within the machine itself.

This approach to guide development, whilst essential, fails to take into account that the majority of the metrology work carried out using CSIs in industrial applications does not take place

within specialised, isolated laboratories. CSIs used in industry are frequently housed with a host of other manufacturing machines in workshops, possibly close to areas where manufacturing work is being carried out [4].

The industrial situation has the result that the measurement data obtained from industrial CSIs can be significantly less accurate than those acquired from CSIs in the NMIs. For example, in the work towards ISO metrological characteristics at the National Physical Laboratory, UK, the measurement noise when measuring an optical flat was in the sub-nanometre range [5]. However, we have found that, with just a small increase in the ambient vibration level, it is easily possible to have measurement noise levels exceeding 20 nm.

A significant component of the measurement noise comes from environmental disturbances in the form of, for example, ground vibration or acoustic noise. These disturbances are observed at a much greater magnitude in an industrial workshop compared to an NMI's laboratory.

At the time of writing, and to the best of the authors' knowledge, there has yet to be any guide published which deals with the issues faced when attempting to obtain areal surface texture data close to the nanometre level in an industrial environment, although some manufacturers have addressed the issue using software and hardware modifications [6]. This paper aims to identify issues introduced by taking measurements in a common working environment. Specific attention is paid to identifying issues causing the increased levels of measurement noise observed in industry and to outlining steps that can easily be taken by technicians or users to reduce the noise levels.

MEASUREMENT NOISE

Measurement noise comprises all the noise signals that add to the output signal of an instrument when the instrument is used in normal operating conditions [7]. Noise can be caused by

both the internal noise of the instrument (for example, electronic instability) and environmental noise (for example, ground vibration, air turbulence, temperature fluctuations). In this study, the measurement noise associated with CSI under industrial conditions will be quantified. Both a subtraction and an averaging method (see [5] for details) will be used to estimate the instrument noise used in industrial conditions. Both the subtraction and the averaging methods use an Sq parameter, which is the root mean square (RMS) value calculated from the departure of heights from a mean reference plane. Two repeated measurements in quick succession are required by the subtraction method. From the two repeated measurements, the surface topography data of one measurement is subtracted from the surface topography data of the other measurement. The subtraction process is used to remove the effect of roughness and the residual flatness error of the surface topography from the noise calculation. The subtraction method combines the surface topography variance of two identical probability distributions of the two measurements and is formulated as:

$$Sq_{noise} = \frac{Sq}{\sqrt{2}} \quad (1)$$

where Sq_{noise} is the estimated measurement noise of the instrument.

In the averaging method, the assumption is made that the measurement noise contribution decreases (converges to a lower value) when averaging two or more surface topography data captured at the same location on the optical flat. The basic idea of the averaging method is that the calculated Sq on the optical flat surface is constituted by the sum of square of the instrument noise (Sq_{noise}) and the optical flat roughness (Sq_{flat}), and is formulated as:

$$Sq = \sqrt{Sq_{flat}^2 + Sq_{noise}^2} \quad (2)$$

The Sq_{noise} will be reduced after n repeated measurements by the square root of n , while the Sq_{flat} is constant (measured at the same location on the optical flat). n measurements are taken and equation (2) gives the RMS of the averaged topography data:

$$Sq_{mean} = \sqrt{Sq_{flat}^2 + \frac{1}{n}Sq_{noise}^2} \quad (3)$$

Hence, the Sq_{noise} can be derived as follows:

$$Sq_{noise} = \sqrt{\frac{Sq_{flat}^2 - Sq_{mean}^2}{1 - 1/n}} \quad (4)$$

Theoretically, both the subtraction and the averaging methods to calculate the measurement noise should have close agreement if all the measured surface topographies have an identical probability distribution; under so-called stationary conditions. The stationary condition of the surface data, obtained over a series of surface acquisitions at the same location on the optical flat, is characterised by constant statistical properties, for example, mean and variance, as they come from the same probability distribution.

Figure 1 shows an example of a measurement noise map calculated by using the subtraction method.

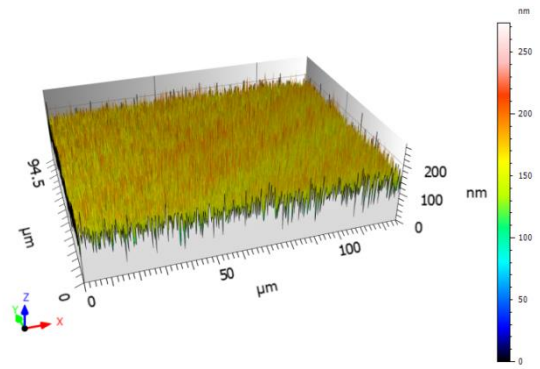


FIGURE 1. Surface after the subtraction method for calculating measurement noise was applied to an optical flat.

In this study, the CSI used is placed under an air conditioning fan blower and is housed in a room with a scanning electron microscope (SEM) instrument with a vacuum compressor. The CSI is mounted on a vibration isolation table. The specification of the vibration isolation table states that the table can isolate low-frequency ground vibration at frequencies less than 4 Hz. The room which houses the CSI instrument is in a manufacturing facility building containing several conventional and non-conventional material cutting machines.

For the measurement noise experiment, a 50× magnification objective lens with a numerical aperture (NA) of 0.55 was used. The objective lens is a Mirau interference objective. The

artefact measured is an optical flat. The lowest measurement speed was applied to vertically scan the optical flat surface through the surface focus position. The measurement noise experiment was carried out in two conditions with the air conditioning fan on and off. The vacuum compressor of the SEM was on in both conditions (it was not possible to switch it off for this work).

RESULTS AND DISCUSSION

Both the subtraction and the averaging methods were applied to estimate the CSI's measurement noise. The results from the averaging method did not converge to a lower noise value. Averaging up to ten measurements of the optical flat surface topography did not reduce the Sq_{noise} . These diverging noise values suggest that the surface data from several measurements do not have an identical probability distribution and are in a non-stationary condition.

The reason for the non-stationary condition could be either due to measuring different locations on the optical surface or due to invalid surface reconstruction by the instrument (i.e. the measured surface topography is different to the real surface topography of the optical flat). As can be seen in figure 2, a sinusoidal (ripple) pattern is observed on the measured surface topography. This ripple pattern is an artificial artefact and is not a representation of the real optical flat surface. The ripple pattern on the measured surface topography suggests that the non-stationary condition is caused by invalid surface reconstruction of the real surface.

The reason of the invalid surface reconstruction is that the environmental noise from the ground and acoustic sources combines into the fringe data while the CSI scans the optical flat and negatively affects the fringe demodulation algorithm of the CSI instrument to reconstruct the surface topography. The environmental noise effects cause invalid features (artefacts) on the measured surface topography and hence induce a non-stationary condition.

Due to the non-stationary property, only the subtraction method is used to estimate the measurement noise. Figure 3 shows the effect of non-stationary surface data to the subtracted surface. In figure 3, a ripple pattern which has a high deviation from the reference surface has been observed. It is worth noting that the ripple pattern is not found in a subtracted surface from

two statistically stationary surfaces as shown in figure 1.

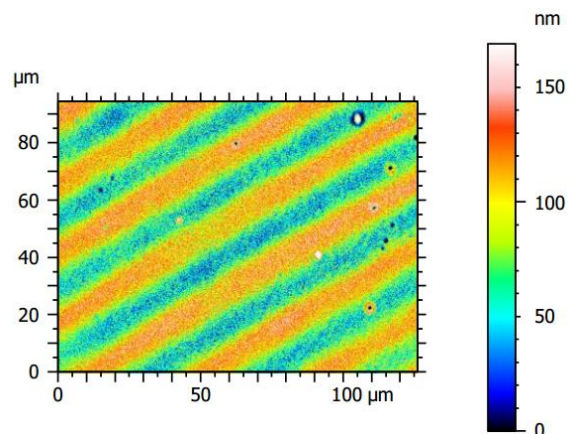


FIGURE 2. The measured surface while the air conditioner's fan blower is on.

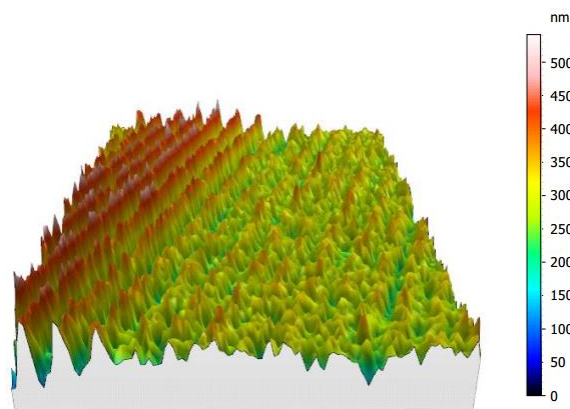


FIGURE 3. Effect of statistically non-stationary surface data to the subtraction between two consecutive surface data.

The measurement noise from the two measurement conditions (air conditioning on and off) shows a significant difference. Eliminating one of the vibration sources (the air conditioning fan) reduces the measurement noise. Table 1 shows the estimated measurement noise for both conditions of the air conditioning fan. The average value of the measurement noise when the air conditioning fan was on is $36.3 \text{ nm} \pm 6 \text{ nm}$ (1σ) and when the air conditioning fan was off is $27 \text{ nm} \pm 2.3 \text{ nm}$. By reducing the vibration source from the air conditioning fan, a 40 % improvement of the measurement noise is obtained. However, the improved noise of 27 nm is still two orders of magnitude higher compared to the noise value of 0.6 nm (a CSI with 50x magnification lens)

obtained in a strict laboratory environment by Giusca et al [4].

TABLE 1. Results of measurement noise from two measurement conditions.

Air conditioner's fan	Measurement noise /nm		
	1	2	3
On	32.4	43.3	33.3
Off	26.7	29.5	24.9

Due to a significant environmental vibration noise contribution, the sinusoidal (ripple) pattern on the surface data becomes apparent. In fact, this sinusoidal pattern is also found in the measurement using phase-shifting interferometry in the presence of vibrations [8]. As per our results, figure 2 shows a significant sinusoidal pattern on the obtained surface data measured when the air conditioning fan was on. A less apparent sinusoidal pattern is observed in figure 4. In figure 4, the surface was measured when the air conditioning fan was switched off. The sinusoidal pattern in figure 2 has a maximum amplitude up to around 100 nm.

To further investigate the vibration sources, an accelerometer sensor was used to measure the vibration during the measurement. The accelerometer sensor was attached to the CSI base table. The CSI's objective lens was set on top of the sensor close to the accelerometer sensor top surface to simulate the real measurement run. The air conditioning fan was switched on and off during the vibration measurement. The signal received from the accelerometer sensor was analysed in the frequency domain to decompose the vibration signal into its single frequency constituents. The frequency spectrum of the measured signal is shown in figure 5. Figure 5 shows the frequency spectrum of the obtained signal recorded from one of the series of vibration measurements.

From figure 5, the frequency spectrum of the measured vibration reveals that the air conditioning fan is not the dominant source of the vibration. The vibration from an air conditioning fan commonly vibrates at frequency around 50 Hz. Because the air conditioning fan is not the dominant vibration source, the vibration noise affecting the measured surface topography is still significant so that the measurement noise is still considered high when the air conditioning fan

was switched off, even though the noise can be improved up to around 40 %. As shown in figure 5, the lower frequency vibration is the dominant vibration source.

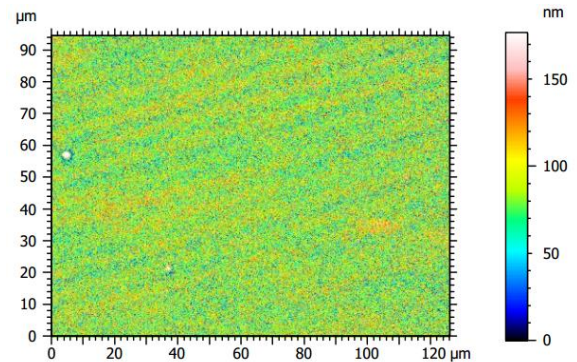


FIGURE 4. The measured surface while the air conditioner's fan blower is off.

To analyse the frequency spectrum, a pick-peaking algorithm, to automatically detect the maximum peak of the signal, was used to find the frequency with the highest amplitude. From a sequence of ten vibration measurements with the air conditioning fan switched on and off, the average frequency having the highest amplitude is $15.1 \text{ Hz} \pm 1 \text{ Hz}$ (1σ). The vacuum compressor of the SEM instrument was working when the vibration was measured. A maximum rotation of 1450 rpm can be performed by the compressor of the SEM instrument. The rotation of the compressor at 1450 rpm corresponds to a 24.17 Hz vibration frequency.

By comparing the frequency with the highest amplitude of the measured vibration signal and the frequency of the vacuum compressor at its maximum rotational speed, a frequency difference of 9 Hz is observed. The vibration frequency difference may be due to the vibration frequency exhibited by the vacuum compressor being attenuated when the vibration of the vacuum compressor propagated to the CSI instrument. The attenuation of the vibration is possibly due to a non-linear damping and isolation effects of the floor concrete and joints through which the vibration propagates. A certain amount of vibration energy dissipates during the travelling of the vibration wave from the vacuum compressor to the CSI. From the comparison of the frequency of compressor motor and the most dominant measured vibration frequency, the compressor motor possibly contributes, to a certain degree, to the vibration noise during the

measurement of the optical flat and causes the increase of measurement noise of the CSI.

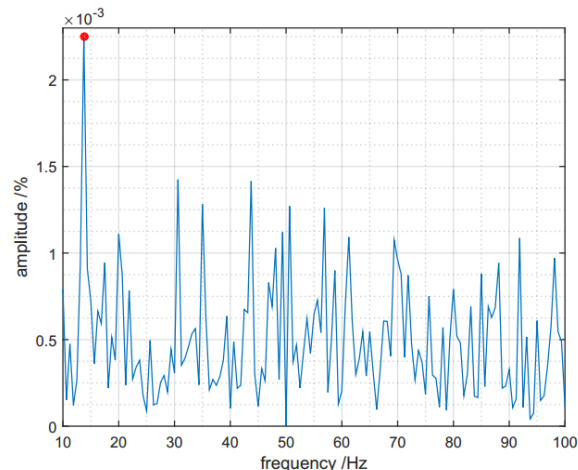


FIGURE 5. The frequency spectrum of the measured vibration signal.

In addition to the vibration noise, the measurement noise of the CSI is significantly affected by the optical flat's surface orientation (surface tilting) with respect to CSI's objective lens. Figure 6 shows the effect of the surface orientation of the estimated measurement noise. As figure 6 suggests, a larger measurement noise is observed when the orientation (tilting) angle of the optical flat increases. It can be argued that the limited numerical aperture (NA) of the objective lens causes a positive correlation between the tilting angle of the optical flat and the estimated measurement noise. The reason is possibly due to the higher tilting angle of the optical flat surface causing some reflected light outside NA limit of the objective lens and causing speckle in the reflected signal to the objective lens [9]. The speckle introduces higher frequency components in the reflected light signals that are outside the bandwidth limit of the CSI [10]. The higher frequency components outside the bandwidth limit will be cut off and cannot pass the optical system. Due to the frequency cut off, some information of the real surface topography is lost.

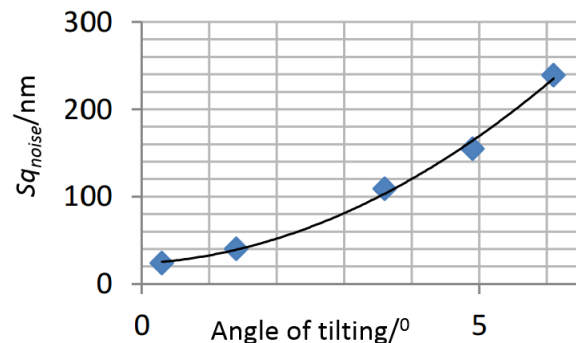


FIGURE 6. The effect of surface tilting to the measurement noise.

The general rules to reduce vibration are by reducing the vibration sources, isolating the vibration sources and/or isolating the instrument. From this study, if low-noise operation is required, a CSI should not be placed together with rotating equipment or another instrument that may generate vibration. Moreover, an air flow system inside the CSI's room should be carefully planned. Finally, the placement of the sample to be measured should be as flat as possible.

CONCLUSION AND FUTURE WORK

This study investigates the measurement noise of a CSI when it is used in an industrial environment. A significantly high measurement noise is observed when the measurement is carried out in the industrial environment as compared to the strictly controlled laboratory environment. The increase of measurement noise is not significant on a rough surface ($Sa > 0.1 \mu\text{m}$) since the sinusoidal pattern noise amplitude observed in the measured surface data due to vibration is around 100 nm. The results suggest that it is important to evaluate the measurement noise of a CSI and other instruments in the actual environment where the instrument will be operated [11]. Continuous improvement, especially by the commercial CSI instrument manufacturers, should be conducted and existing knowledge related to vibration noise on measurements should be published more openly (for example, as can be found in [12]).

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POLYNOMIAL-MATRIX METHOD FOR HIGH-PRECISION ANALYSIS OF ROTATIONAL SHEARING INTERFEROGRAMS

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INTRODUCTION

An interferometer is often used to measure the shape of a lens or a mirror. Interferograms obtained via a Fizeau interferometer, Twyman-Green interferometer, etc provide a contour map of a wave front under test of the measurement surface.

On the other hand, shearing interferograms show the difference between the wave front under test and a sheared wave front along the shearing direction. In other words, a shearing interferogram shows the inclination of the wave front under test. Therefore, the shape of the wave front under test is reconstructed by means of the shearing interferogram. In previous work, we proposed a Polynomial-Matrix method for high-precision analysis of lateral shearing interferograms [1].

In this paper, we propose a method for applying the Polynomial-Matrix method to the analysis of rotational shearing interferograms.

ROTATIONAL SHEARING INTERFEROGRAMS

In rotational shearing interferometry [2], the intensity $I(\rho, \theta)$ of an interferogram can be expressed in terms of polar coordinates ρ and θ , as follows:

$$I(\rho, \theta) = \frac{1}{2} \left[1 + \cos \left\{ \frac{2\pi}{\lambda} \Delta W(\rho, \theta, \phi) \right\} \right] \quad (1)$$

where λ is the wavelength of the light source, ρ is the radius, θ is the angle, ϕ is the amount of shear, and ΔW is difference between the wave front and sheared wave front, as given by:

$$\begin{aligned} \Delta W(\rho, \theta, \phi) \\ = W\left(\rho, \theta + \frac{\phi}{2}\right) - W(\rho, \theta - \phi/2), \end{aligned} \quad (2)$$

where W is a function of the wave front under test. Equation (3) is then obtained by dividing both sides of equation (2) by ϕ :

$$\frac{\Delta W(\rho, \theta, \phi)}{\phi} = \frac{W(\rho, \theta + \phi) - W(\rho, \theta)}{\phi}. \quad (3)$$

In the limit as $\phi \rightarrow 0$, we have:

$$\lim_{\phi \rightarrow 0} \frac{W(\rho, \theta + \phi) - W(\rho, \theta)}{\phi} = \frac{\partial W(\rho, \theta)}{\partial \theta}. \quad (4)$$

Equation (4) shows the definition of a derivative. However, when $\phi = 0$, an interferogram is not obtained by a rotational shearing interferometer. Accordingly, ϕ is a finite value and Equation (4) is not satisfied. As a result, a rotational shearing interferogram shows the difference between the wave front under test and the sheared wave front along the shearing direction, as follows:

$$\frac{W(\rho, \theta + \phi) - W(\rho, \theta)}{\phi} \cong \frac{\partial W(\rho, \theta)}{\partial \theta}. \quad (5)$$

PROPOSED METHOD

Because the magnitude of the shear has a finite value, a rotational shearing interferogram does not obtain the differential of a wave front, but rather the difference between the wave front and the sheared wave front. Therefore, when the obtained interferograms are analyzed via the

integration method, the analyzed wave front will contain an error. In this section, we show that this error is systematic, and a compensating method for this systematic error is proposed.

Systematic error

The resulting error may be investigated using Zernike polynomials. For example, the difference of the wave front, $\Delta W_2(\rho, \theta)$ obtained by a rotational shearing interferometer is expressed in terms of the second order Zernike polynomial, as follows:

$$\begin{aligned}\Delta W_2(\rho, \theta, \phi) &= W_2(\rho, \theta + \phi/2) \\ &\quad - W_2(\rho, \theta - \phi/2) \\ &= \rho \sin(\theta + \phi/2) \\ &\quad - \rho \sin(\theta - \phi/2) \\ &= 2 \sin \frac{\phi}{2} \times \rho \cos \theta.\end{aligned}\quad (6)$$

Equation (7) is obtained by integrating both sides of equation (6):

$$\begin{aligned}W_{2error}(\rho, \theta, \phi) &= \int \frac{\Delta W_2(\rho, \theta, \phi)}{\phi} d\theta \\ &= \int \left(\frac{2 \sin \frac{\phi}{2} \times \rho \cos \theta}{\phi} \right) d\theta \\ &= \frac{\sin \frac{\phi}{2}}{\frac{\phi}{2}} \times \rho \sin \theta\end{aligned}\quad (7)$$

An analysis error is defined by $W_{2error}(\rho, \theta, \phi)/W_2(\rho, \theta)$, as follows:

$$\frac{W_{2error}(\rho, \theta, \phi)}{W_2(\rho, \theta)} = \frac{\sin \frac{\phi}{2}}{\frac{\phi}{2}}\quad (8)$$

Similarly, in Table 1, the analysis errors are listed up to order $i = 16$ of the Zernike polynomials. The relationship between i , n , and m can be described as:

$$i = \frac{n(n+1)}{2} + m + 1.\quad (9)$$

A general term of the analysis error can be read from Table 1, as follows.

$$\begin{aligned}\frac{W_{ierror}(\rho, \theta, \phi)}{W_i(\rho, \theta)} \\ = (1 - \delta_{n-2m,0}) \text{sinc} \frac{|n-2m|\phi}{2},\end{aligned}\quad (10)$$

where δ is Kronecker delta and sinc is non-regular sinc function. Accordingly, the error is the systematic error.

Compensating method

In the Polynomial-Matrix method, the Zernike polynomial and matrix operations are used for analysis of the interferograms. For the matrix operations, we consider the matrix T and the vectors A and B . For example, the components of vector A are defined in the terms of the Zernike polynomials, up to the 5th order, as follows.

$$A = \begin{bmatrix} 1 \\ \rho \cos(\theta) \\ \rho \sin(\theta) \\ \rho^2 \cos(2\theta) \\ -1 + 2\rho^2 \end{bmatrix}\quad (11)$$

The components of vector B are defined as the integration of the difference obtained by shearing the Zernike polynomials, as follows:

$$B = \begin{bmatrix} 0 \\ \frac{\rho \cos(\theta) \sin(\frac{\phi}{2})}{\frac{\phi}{2}} \\ \frac{\rho \sin(\theta) \sin(\frac{\phi}{2})}{\frac{\phi}{2}} \\ \frac{\rho^2 \cos(2\theta) \sin(\phi)}{\phi} \\ 0 \end{bmatrix}\quad (12)$$

As the difference is not the differential, vector A is not consistent with vector B , and a component of vector B includes the analysis error. The relationship between vectors A and B can be expressed as follows:

$$B = TA.\quad (13)$$

Therefore,

$$A = T^{-1}B\quad (14)$$

where T is a square singular matrix, as follows.

TABLE 1. The analysis errors up to order $i = 16$ of the Zernike polynomials.

i	n	m	$W_i(\rho, \theta)$	$W_{i_{error}}(\rho, \theta, \phi)$	$\frac{W_{i_{error}}(\rho, \theta, \phi)}{W_i(\rho, \theta)}$
1	0	0	1	0	0
2	1	0	$\rho \sin(\theta)$	$\frac{\rho \sin(\theta) \sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$	$\frac{\sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$
3	1	1	$\rho \cos(\theta)$	$\frac{\rho \cos(\theta) \sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$	$\frac{\sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$
4	2	0	$\rho^2 \sin(2\theta)$	$\frac{\rho^2 \sin(2\theta) \sin(\phi)}{\phi}$	$\frac{\sin(\phi)}{\phi}$
5	2	1	$-1 + 2\rho^2$	0	0
6	2	2	$\rho^2 \cos(2\theta)$	$\frac{\rho^2 \cos(2\theta) \sin(\phi)}{\phi}$	$\frac{\sin(\phi)}{\phi}$
7	3	0	$\rho^3 \sin(3\theta)$	$\frac{\rho^3 \sin(3\theta) \sin\left(\frac{3\phi}{2}\right)}{\frac{3\phi}{2}}$	$\frac{\sin\left(\frac{3\phi}{2}\right)}{\frac{3\phi}{2}}$
8	3	1	$(-2\rho + 3\rho^3) \sin(\theta)$	$\frac{(-2\rho + 3\rho^3) \sin(\theta) \sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$	$\frac{\sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$
9	3	2	$(-2\rho + 3\rho^3) \cos(\theta)$	$\frac{(-2\rho + 3\rho^3) \cos(\theta) \sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$	$\frac{\sin\left(\frac{\phi}{2}\right)}{\frac{\phi}{2}}$
10	3	3	$\rho^3 \cos(3\theta)$	$\frac{\rho^3 \cos(3\theta) \sin\left(\frac{3\phi}{2}\right)}{\frac{3\phi}{2}}$	$\frac{\sin\left(\frac{3\phi}{2}\right)}{\frac{3\phi}{2}}$
11	4	0	$\rho^4 \sin(4\theta)$	$\frac{\rho^4 \sin(4\theta) \sin(2\phi)}{2\phi}$	$\frac{\sin(2\phi)}{2\phi}$
12	4	1	$(-3\rho^2 + 4\rho^4) \sin(2\theta)$	$\frac{(-3\rho^2 + 4\rho^4) \sin(2\theta) \sin(\phi)}{\phi}$	$\frac{\sin(\phi)}{\phi}$
13	4	2	$1 - 6\rho^2 + 6\rho^4$	0	0
14	4	3	$(-3\rho^2 + 4\rho^4) \cos(2\theta)$	$\frac{(-3\rho^2 + 4\rho^4) \cos(2\theta) \sin(\phi)}{\phi}$	$\frac{\sin(\phi)}{\phi}$
15	4	4	$\rho^4 \cos(4\theta)$	$\frac{\rho^4 \cos(4\theta) \sin(2\phi)}{2\phi}$	$\frac{\sin(2\phi)}{2\phi}$
16	5	0	$\rho^5 \sin(5\theta)$	$\frac{\rho^5 \sin(5\theta) \sin\left(\frac{5\phi}{2}\right)}{\frac{5\phi}{2}}$	$\frac{\sin\left(\frac{5\phi}{2}\right)}{\frac{5\phi}{2}}$

$$T = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{\sin(\frac{\phi}{2})}{\frac{\phi}{2}} & 0 & 0 & 0 \\ 0 & 0 & \frac{\sin(\frac{\phi}{2})}{\frac{\phi}{2}} & 0 & 0 \\ 0 & 0 & 0 & \frac{\sin(\phi)}{\phi} & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (15)$$

As a singular matrix does not have an inverse, the square-enclosed zero value among the diagonal elements is replaced by unity:

$$M = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & \frac{\sin(\frac{\phi}{2})}{\frac{\phi}{2}} & 0 & 0 & 0 \\ 0 & 0 & \frac{\sin(\frac{\phi}{2})}{\frac{\phi}{2}} & 0 & 0 \\ 0 & 0 & 0 & \frac{\sin(\phi)}{\phi} & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \quad (16)$$

in which case:

$$M^{-1} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & \frac{\phi}{2 \sin(\frac{\phi}{2})} & 0 & 0 & 0 \\ 0 & 0 & \frac{\phi}{2 \sin(\frac{\phi}{2})} & 0 & 0 \\ 0 & 0 & 0 & \frac{\phi}{\sin(\phi)} & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}. \quad (17)$$

As a result, it is possible to obtain vector A without error via the simple expression:

$$A = M^{-1}B. \quad (18)$$

SIMULATIONS

Simulations were carried out under the following conditions. The test surface was first defined as follows:

$$W(\rho, \theta) = \rho \cos \theta + \rho^2 \sin 2\theta + \rho(3\rho^2 - 2) \cos \theta + \rho^3 \cos 3\theta \quad (19)$$

The shear amount, ϕ was set from 0 to 0.25% and the analysis radius ρ was set at 0.5.

Figure 1 shows the results obtained by both the conventional integration method and the proposed Polynomial-Matrix method. The blue squares and the red circles respectively show the conventional integral method and the proposed Polynomial-Matrix method. The analysis error of the proposed method is very small, regardless of the shear amount.

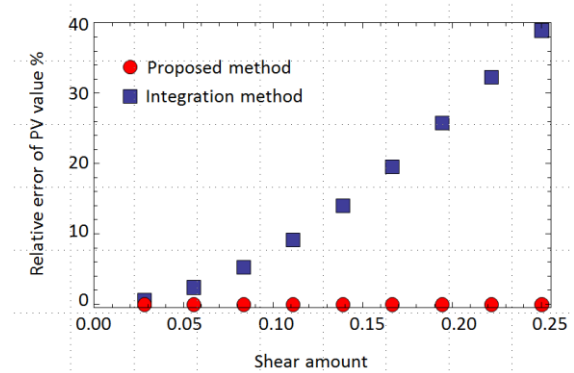


FIGURE 1. Relationship between the analysis error and amount of shear.

CONCLUSIONS

In this paper, a new method was proposed to analyze rotational shearing interferograms without error. The validity of the proposed method was then confirmed via simulation.

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Design and Fabrication of a Multi-layer Capacitive Transducer for Normal and Shear Foot Plantar Pressure Measurements

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INTRODUCTION

Plantar pressure distribution is an important research field for various biomechanics fields such as rehabilitation medicine, gait analysis, anatomical inspection, patients' diabetes status forecasting, and neuromuscular examination [1][2][3][4]. Stress distribution data can also be used to guide sportsmen to improve posture and hence to reduce some injuries [5]; for shoe manufacturing industry, engineers could customize the design of different products based on the sole stress data [6]; with the increasing number of running and working-out population, normal and shear force distribution data could be used to rectify posture and lower the risk of injuries.

The plantar stresses can be divided into two components: normal stress (i.e., vertical to the sole surface), and shear stress (i.e., parallel to the sole surface). Various methods to obtain pressure measurements have been reported in the literature. JJ Wertsch and JG Webster [7] manufactured some effective pressure sensor installed on an insole; the design measured normal pressure, but not the shear pressure. E.J Heywood et al.[8] manufactured a platform to measure both normal and shear forces; nonetheless the design was rather large and not suitable to be installed in a shoe. J.A Dobrzynska [9], Hyung-Kew Lee [10] and Peter Roberts [11] proposed a small and flexible sensor for shear force measurement, but did not overcome some disadvantages such as the limited measuring force range, the complicated fabrication technique, and the relatively weak response signal compared to the proposed technique.

In this manuscript, we proposed using a relatively simple manufacturing technique based on known manufacturing technologies to make relatively small transducers capable of measuring shear and normal stresses concurrently.

PROPOSED DESIGN

Sensing technology

Various sensing technologies are available to build pressure transducers, such as strain gauge, optical interferometer, piezoelectric sensor, capacitive sensor, and magneto-resistor sensor etc. [12]. In this work we chose capacitive sensors due to their relatively simple working principle and structure, and their ability to measure dynamic forces [13].

Dielectric material

For a thin-film capacitor with an external load applied in a direction parallel to the thickness, the thickness of dielectrics change, causing the capacitance to change as well. The transducer's static capacitance C_0 can be expressed as:

$$C_0 = \frac{\epsilon_r \epsilon_0 S}{d_0} \quad (1)$$

ϵ_r is the material's relative dielectric constant, ϵ_0 is the dielectric constant of air, S is the electrode area, d_0 is the static thickness of the film. As the film is compressed the thickness of dielectrics changes and the corresponding capacitance can be calculated as:

$$C = C_0 + \Delta C = \frac{\epsilon_r \epsilon_0 S}{d_0 - \Delta d} = C_0 \times \frac{1}{1 - \frac{\Delta d}{d_0}} \quad (2)$$

By applying a Fourier expansion (assuming $\Delta d \ll d_0$), equation (2) can be approximated as:

$$\Delta C \approx C_0 \times \frac{\Delta d}{d_0} \quad (3)$$

Ignoring the Poisson effect, the strain, $\frac{\Delta d}{d_0}$ is equal to $\frac{\Delta P}{E}$, where ΔP is the pressure applied and E the dielectrics' Young's Modulus, equation (3) can be written as:

$$\Delta C \approx \Delta P \times \frac{\epsilon_0 S}{d_0} \times \frac{\epsilon_r}{E} \quad (4)$$

The change in capacitance is proportional to $\epsilon_r / (d_0 E)$. At the same time, this dielectric shall be as thin as possible. In addition to considering the electrical performance, the selected dielectric material should have low hysteresis during the walking/running period and low creep rate at the

frequency considered. Based on the above requirements, various elastomers and plastics were selected as candidates.

TABLE 1. Typical Performance of PDMS and PI [14][15]

Material	Young's Modulus (Pa)	Dielectric Constant	$\frac{\epsilon_r}{E}$	Min Thickness (μm)
PDMS	2.0×10^6	2.5	1.25×10^{-6}	20
PI	2.5×10^9	3.4	1.36×10^{-9}	8

Among the materials considered, Polydimethylsiloxane (PDMS) was chosen due to its high ϵ_r/E ratio. In addition, PDMS can be manufactured in thin films (table 1). Besides, PDMS is can withstand over 1MPa in compression [14] which is below the expected range for plantar pressure. Polyimide, (PI), is also taken into consideration since it is one of the most-widely used polymer in the electronics industry (e.g., as the base material for flexible printed circuits,) it is very durable, and can be manufactured extremely thin.

Transducer's Structure

The proposed transducer can be divided into three basic components. Figure 1 illustrates the main structure. From top to bottom, the transducer comprises (1) a stiff cover with plungers pressing on the capacitors and a bump, (2) three capacitor stacks with PDMS fixtures, and (3) a printed circuit board base. The bump is connected with the plungers and allows the generation of torque when shear forces are applied. Whereas when a normal force is loaded, all three capacitors (C_1 , C_2 and C_3) are compressed simultaneously. Since there are only three orthogonal components of the applied force, 3 capacitors are appropriate to fully resolve for the applied forces.

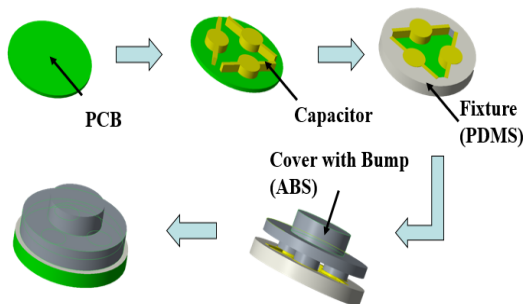


FIGURE 1. Schematic of main body structure of the whole transducer

Stack Capacitors

To increase the response of the transducer, each capacitor is made of a stack of thin film capacitors. Figure 2 shows the detail construction of each individual capacitors-stack. Each capacitor unit consists of three components: the flexible printed circuit (i.e., the electrodes), the dielectrics films (PDMS and PI), and the wires to connect the electrodes in parallel. This multi-layer sandwich structure connects each thin film capacitor in parallel and increases the overall unit's capacitance. The FPCs are disposed in a staggered fashion to minimize the chances of shorting. For the same purpose, FPCs are copper-clad on one side only and the result is PI is also a part of the dielectrics' stack. The contribution of the PI film to the overall capacitance was found to be negligible.

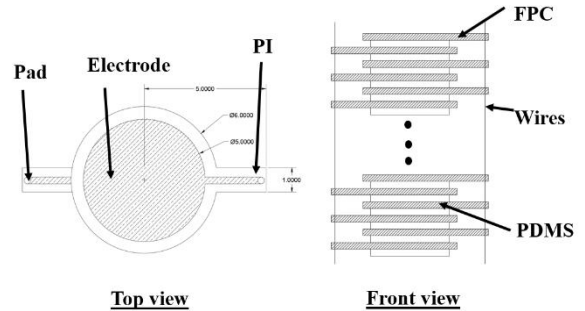


FIGURE 2. The multi-layer structure of the capacitor units

VALIDATION

Analytical Model

A simplified, static analytical model to describe the working principle of the transducer is introduced in this section. A schematic representation of the side and top view of the transducer is also shown in Figure 3. Assuming a normal force, N , and two orthogonal components of the shear force, T_x and T_y , are applied at the center of the bump, the normal pressure on each stack can be computed as:

$$\begin{cases} P_1 = T_y \left[\frac{H}{l(1 + \sin \alpha)} \right] + \frac{N}{3} \\ P_2 = -T_x \left(\frac{H}{2l \cos \alpha} \right) - T_y \left[\frac{H}{2l(1 + \sin \alpha)} \right] + \frac{N}{3} \\ P_3 = T_x \left(\frac{H}{2l \cos \alpha} \right) - T_y \left[\frac{H}{2l(1 + \sin \alpha)} \right] + \frac{N}{3} \end{cases} \quad (5)$$

In the above equations, H stands for the height of bump from the PCB, l stands for the distance between the center of the PCB and the center of each stack, α is the angle between the horizontal line and the transducers (typically, 30°).

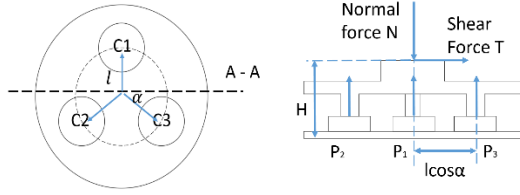


FIGURE 3. Schematic diagram of analytical model (top view on the left; side view on the right)

Combining these equations to equation (6), the output capacitance of each transducer stack is obtained as:

$$\begin{cases} C_1 = \frac{\epsilon_r \epsilon_0 S E}{d_0} \times \frac{1}{E - T_y \left[\frac{H}{l(1 + \sin \alpha)} \right] - \frac{N}{3}} \\ C_2 = \frac{\epsilon_r \epsilon_0 S E}{d_0} \times \frac{1}{E + T_x \left(\frac{H}{2l \cos \alpha} \right) + T_y \left[\frac{H}{2l(1 + \sin \alpha)} \right] - \frac{N}{3}} \\ C_3 = \frac{\epsilon_r \epsilon_0 S E}{d_0} \times \frac{1}{E - T_x \left(\frac{H}{2l \cos \alpha} \right) + T_y \left[\frac{H}{2l(1 + \sin \alpha)} \right] - \frac{N}{3}} \end{cases} \quad (6)$$

The three equations generated by three capacitors, can be used to solve for the three components of the force. Also, to avoid tension in each stack, the value of H and l shall be carefully selected. Figure 4 is the calculation obtained based on equation 6 based on the expected range of normal pressure and shear stress 0-500kPa and 0-100kPa respectively [16]. Hence based on $T/N < 0.2$, the length l was chosen as 6mm, hence the total height of the transducer (including the bump) was taken as 10mm.

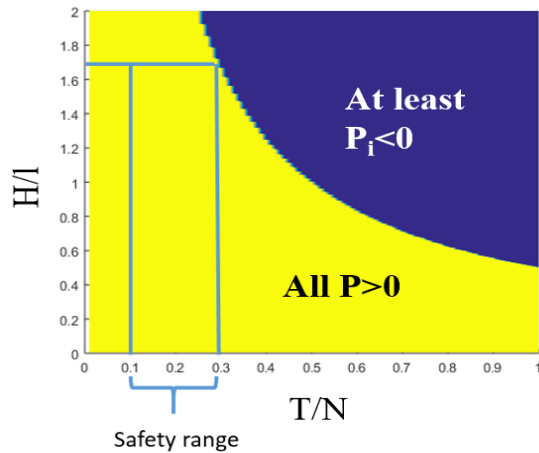


FIGURE 4. Picture of the tension area checking diagram based on equations (5)

Numerical Simulation

Since the analytical model is not accurate enough to accurately predict the transducer's output,

finite element analysis model is used to estimate the response of the transducer. The simulation is carried out within a standard commercial package, COMSOL® Multiphysics. 3D solid stress-strain mode with large deformation is used to simulate the response of the assembly when normal and shear loads were applied. All materials are assumed isotropic. The Young's modulus of capacitors is estimated as 9.5GPa (assuming PI accounts for 40% and PDMS accounts for 60%). Figure 5 illustrates the boundary conditions for the case considered.

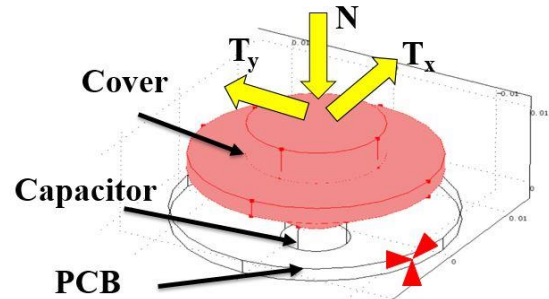
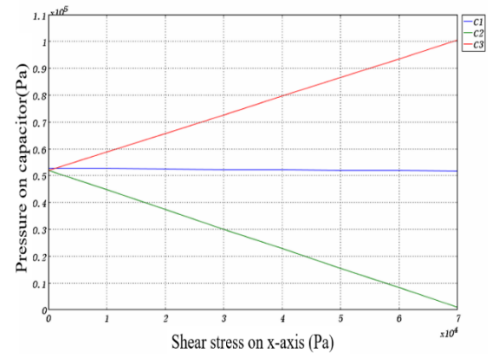
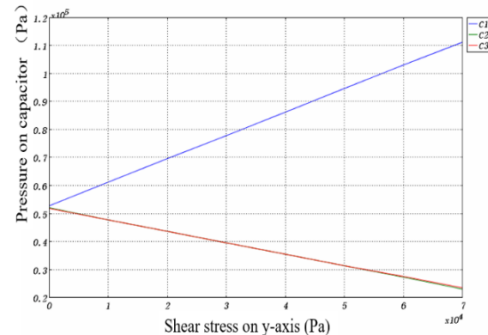


FIGURE 5. Boundary conditions: bottom fixed, force on the top, contact pairs between the cover and capacitors, others are free

And Figure 6 shows pressures on three capacitors with different input shear forces under 200kPa normal force on the bump.



(a)



(b)

Figure 6 Shear stress simulation plots under 200kPa normal pressure: (a) stress on x-axis, (b) stress on y-axis

The simulation results are consistent with the analytical results in terms of behavioral trends. Because of the multi-layer structure, the static capacitance, C_0 , of each transducer could be as large as 95pF (20 PDMS layers, each approximately 65 μm thick), which is nearly 50 times larger than the values reported by other authors [10]. This large output signal is expected to significantly improve the signal-to-noise ratio compared to previous designs. The number of PDMS films layers could also be increased to further amplify the signal if required.

FABRICATION

Capacitors

The dielectric PDMS film is made through spin coating process using a spin coater KW-4A manufactured by Chinese Academy of Sciences. To make a PDMS film approximately 65 μm thick, the rotational speed was set at 1000rpm for 1min. Next, the PDMS film was cured at 80 $^{\circ}\text{C}$ for 1 hour and disk shaped electrodes were made using a circular steel punch. Copper-clad electrodes deposited on a flexible PI substrate are alternated with the PDMS film disks during assembly (Figure 2). To keep the film capacitor in parallel, each lead terminal is connected to every other film as also shown schematically in Figure 2. A sample transducer as manufactured is also shown in Figure 7.

Cover and mold

To simplify the manufacturing process and keep the cost low, the cover of the transducer and the molds for the PDMS protective shell are fabricated through 3D printing technology. 3D printing performed by a standard 3D printer (Winbo WBFDM463131) with accuracy of $\pm 100 \mu\text{m}$. The material used by the printer is Acrylonitrile Butadiene Styrene (ABS).

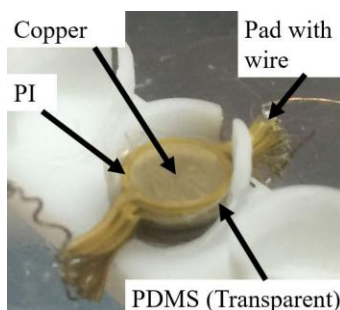


Figure 7 One capacitor stack illustration

Assembly

Together with the PCB from local vendor, capacitors are assembled together with external scaffolding (cover and bottom). All components shall be sealed and protected to prevent moisture ingress. An image of the prototype (without PDMS seal) is shown in Figure 8.

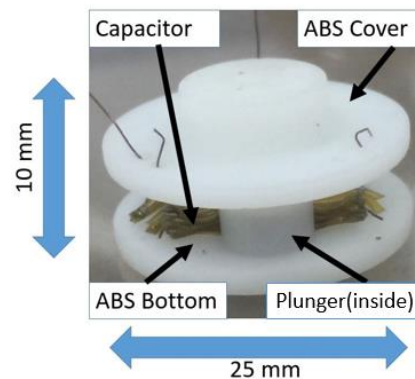


Figure 8 Transducer prototype without PDMS seal

TRANSDUCER'S CHARACTERIZATION

Before the testing for the whole device, some basic characteristics of the capacitor-stack were verified, such as the static capacitance, the change in capacitance versus applied load, the viscoelastic response of the material, and the capacitance hysteresis curve. The results of the measured capacitance versus compressive normal force applied by MTS in the 0–600kPa range are shown in Figure 9. The pressure range was selected based on values found on the literature [16] for dynamic plantar pressure. Sensor loading and unloading characteristics of three capacitors are obtained using a universal loading machine (MTS C45.504 100-230-707) and a loading rate of 0.1 mm/minute. The minimum recorded capacitance is approximately 90pF and the maximum capacitance as 104pF. It was found that at lower pressure (0-50kPa), the sensitivity was about 0.1nF/MPa while for larger stress, the sensitivity was about 4.5pF/MPa. Figure 9 shows a plot for the capacitance versus the applied pressure for three transducer stacks (20 layers each). Although some hysteresis was observed at the loading rate considered (0.1 mm/min), its effect is not very significant and it is expected to be less significant at higher loading rates.

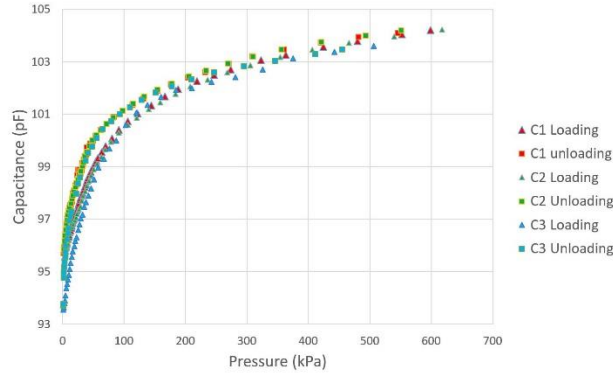


Figure 9 Measurement results upon applying compressive normal force to three capacitor stacks (loading rate: 0.1mm/min).

Figure 10 shows the force–displacement hysteresis curve for a displacement of 0 to 0.210mm and a loading rate of 0.1 mm/min. The three hysteresis curves appear to be repeatable and retracing themselves, with variation less than 1% among the three cycles.

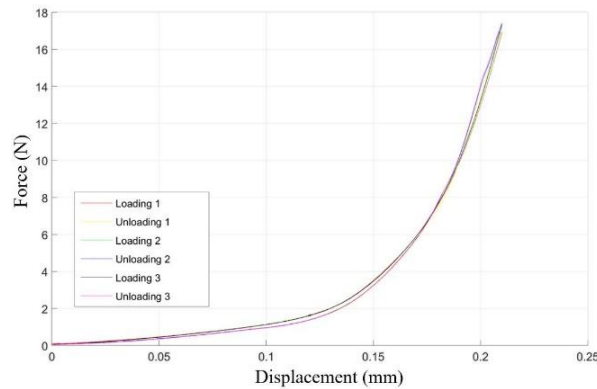


Figure 10 Measurement results upon applying compressive normal force to one capacitor stack with same loading and unloading cycles (rate: 0.1mm/min).

Preliminary tests on the viscoelasticity of the capacitor were carried out by ‘instantons unloading’ of the sample (see, Figure 11). To verify the time constants of the composite capacitor, a linear standard model was used to fit the transducer’s response. For a pressure of 250kPa it was found that the equation below fit the data reasonably well:

$$C = 5.6 * e^{-\frac{t}{20.7}} + 12.278 * e^{-\frac{t}{0.672}} + 81.96 \quad (7)$$

Based on this fitting result, the two time constants are approximately 20.7s and 0.672s respectively although the smaller time constant accounts for

the larger signal amplitude. Since the largest contribution to the response corresponds to the 0.672s time constant, some signal compensation will be necessary as the gait cycle is expected to be in the 1s range [17].

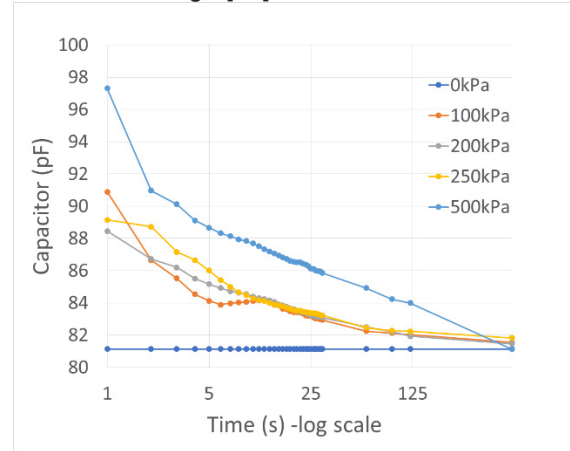


Figure 11 Measurement results upon applying compressive normal force to one Capacitor with same loading and unloading cycles (rate: 0.1mm/min).

CONCLUSIONS

In this paper we introduced the use of low-cost, high response multi-layer capacitive transducer for normal and shear force measurements in plantar surface. PDMS was chosen for fabricating capacitor units by spin coating due to its relatively low Young modulus and reasonable dielectric constant. The capacitor units had multi-stack structure which helped to increase the initial capacitance and the signal-to-noise ratio. A novel design consisting of a sandwich-like structure formed by three capacitor units and loading bump was made to measure 3-axis dimensional force.

In this work we showed that by spin coating and FPC fabrication, a compact capacitance unit with 20 layers and only 1.6 millimeter thickness could be manufactured. A Stacking method was applied to increase the signal output (approximately 50 times larger than the values reported by H-K. Lee [10]) without sacrificing size and measurement range.

Both experimental and numerical techniques have been implemented to verify the validity of the performance of such transducer prototype. This kind of multi-layer structure based capacitive transducer show high potential not only for measuring the plantar force distribution, but also for application such as artificial skin for robot or artificial limb. More layers and different size will

be performed in the future using similar sensor. Finally, a more compact version of the transducer with embedded electronics is planned for the future.

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COMPUTATIONAL DENSITY ESTIMATIONS OF A COORDINATE MEASURING MACHINE ARTIFACT

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INTRODUCTION

Density estimations can be determined in a number of ways but a straightforward approach is volume displacement in a liquid bath, commonly known as Archimedes Principle [1], [2]. The current density measurement process for Pit Manufacturing at Los Alamos National Laboratory (LANL) uses bromobenzene (C_6H_5Br) as the liquid for submersion of nuclear workpieces in the density determination. Bromobenzene is highly-toxic which can cause liver and nervous system damage if consumed [3]. On top of the dangers of using such a liquid and the inability of disposing, storage is a substantial cost thus the need for alternative methods in workpiece density determination.

A proposed alternative is to numerically estimate the workpiece volume using analytical formulae [4] and/or numerical algorithms [5] on readily available dimensional measurement data. Multiple workpieces have been measured using multiple measurement instruments such as rotary contour gauges (RCGs) and traditional coordinate measuring machines (CMMs). This approach is an extension of an in-house quality check which uses analytical models of well-known geometries [6], [7] to estimate the volume of the workpiece. The goal of the research is to test and develop volume algorithms, as well as estimate uncertainty values, for determining density calculations of curved workpieces using dimensional measurements [8].

BACKGROUND

Workpiece density is an important metric in nuclear operations at LANL yet the issue with the storage and disposal of radioactive materials is ongoing. Alternative methods are being investigated to alleviate this issue by utilizing legacy data. In doing so, the need for volume estimations using liquid baths is no longer a necessity, thus eliminating the potential of hazardous events from occurring as well as

minimizing costs associated with the current storage protocol.

Upon completion of the density determination process, the workpiece is dimensionally measured using an RCG or CMM, where a high-density data set is collected. For these curved workpieces, outer and inner contour measurements are collected where the RCG is a direct measurement of the wall thickness, whereas the CMM is an indirect measurement of the wall thickness. As a result of these measurements, it is assumed that the workpiece volume can be estimated using analytical and/or numerical techniques.

After the volume and mass of a workpiece are determined, the density of the workpiece is a simple calculation. The density model is shown below in Eq. 1,

$$\rho = \frac{m}{V} \quad (1)$$

where m is the mass of the workpiece and V is the volume of the workpiece.

Workpiece

For testing, an unclassified representative workpiece is utilized for the determination of the mass and volume. The unclassified artifact is commonly used throughout the Nuclear Weapons Complex (NWC) for testing out new processes as well round-robin testing between the different national laboratories that make up the NWC.

A CAD model of the workpiece is shown in FIGURE 1, where the outside geometry is the combination of a sphere and cone, while the inside geometry is a sphere and a cylinder. This is important to note as the analytical models are derived as a result of.

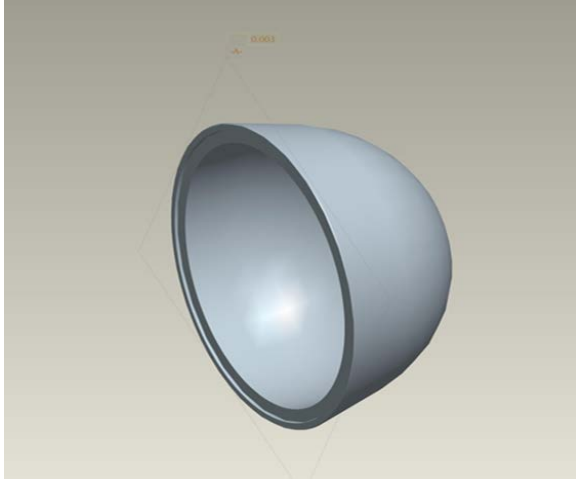


FIGURE 1. Unclassified CMM artifact.

Analytical Modeling

The unclassified CMM artifact definition is divided into both inner and outer contours where each contour is a combination of known prismatic geometries [4]. The treatment of the analytical models will be modeled as combinations of a line and circle in 2D but rotated about a central axis for the 3D representation.

The analytical definition of the inner profile is drawn below in FIGURE 2 where the workpiece center is $(x, y) = (0, 0)$ and the center of the circle is $(x_c, y_c) = (x_0, 0)$. It should be noted that Datum A is at the edge of the inner geometry and not at the origin. The x-axis is the axis of rotation.

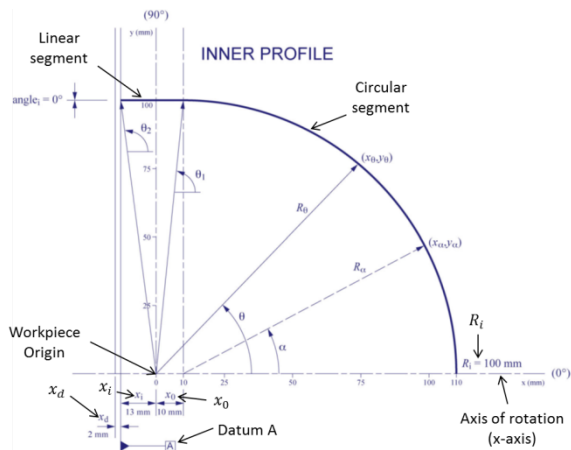


FIGURE 2. Inner profile of CMM artifact, [4].

Calculating the volume of the inner profile is done by assuming that the geometries, when rotated around the x-axis, are a quarter-sphere

(circle) and a cylinder (line). The radii for both geometries are R_i and the thickness of the cylinder is the distance from the workpiece center to the edge. Hence, the volume models are

$$V_{hemi} = \frac{2\pi}{3} R_i^3 \quad (2)$$

$$V_{cylinder} = \pi R_i^2 (x_o + |x_i| + |x_d|) \quad (3)$$

where the volume of the inner contour is simply $V_{inner} = V_{hemi} + V_{cylinder}$.

For the outer profile, the analytical definition is more complicated. The volume is made up of two coaxial cones, where only one cone is shown in FIGURE 3, and a segment of a sphere. To get the additional cone, the straight-line section of the profile is extended until it meets the x-axis ($y = 0$). This will be known as cone 1, where cone 2 is the straight-line segment in FIGURE 3 rotated about the x-axis.

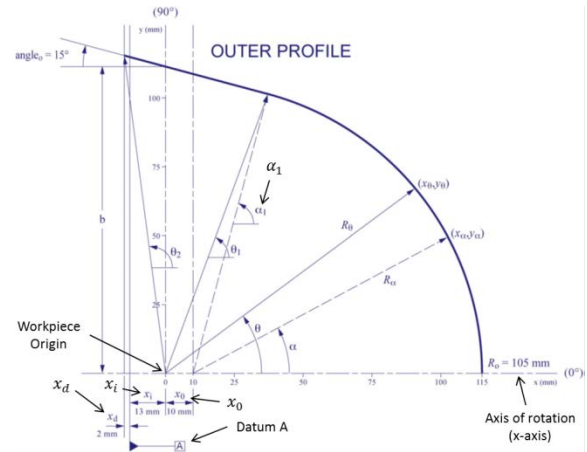


FIGURE 3: Outer profile of CMM artifact, [4].

Hence, the volume models of the three sections are

$$V_{cone1} = \frac{\pi}{3} y_{sharp}^2 h_{c1} \quad (4)$$

$$V_{cone2} = \frac{\pi}{3} y_{a1}^2 h_{c2} \quad (5)$$

$$V_{cap} = \frac{\pi}{3} h_1^2 (3R_0 - h_1) \quad (6).$$

$$y_{sharp} = y_{\alpha_1} + \Delta y' \quad (7)$$

$x_{\alpha_1} = \sqrt{R_0/(1 + \tan^2(\alpha_1))}$, $y_{\alpha_1} = x_{\alpha_1} \tan(\alpha_1)$,
 $x_e = x_0 - x_{\alpha_1} - \Delta x'$, $\Delta x' = -\Delta y'/\tan(\pi/12)$ and
 $\Delta y' = -\tan(\pi/12)(x_{\alpha_1} + |x_d| + |x_i| + |x_0|)$.
 Hence the volume calculation for the outer
 contour is simply $V_{outer} = V_{cone_1} - V_{cone_2} + V_{cap}$.

The figure illustrates the rotation of a body about a fixed axis. On the left, a coordinate system with a vertical \$y\$-axis (in mm) and a horizontal \$x\$-axis (in mm) is shown. A point initially at \$(0, 100)\$ on the \$y\$-axis rotates through an angle \$\theta_1\$ around a pivot point located at \$(0, 100)\$. The radius of rotation is given as \$R_o = 105\text{ mm}\$. The new position of the point is labeled \$(x_d, y)\$. Below the \$x\$-axis, distances are marked: \$X_{d1} = 13\text{ mm}\$, \$X_{d2} = 10\text{ mm}\$, and \$X_{d3} \approx 27.176000\text{ mm}\$. An arrow indicates the direction of rotation.

On the right, the resulting shape after rotation is shown. It is a shaded area bounded by a curved line and straight segments. Key points on its boundary are labeled: \$(0, b \approx 111.383491)\$, \$(-13, 110)\$, \$(-15, 110)\$, and \$(-13, 100)\$. A note specifies "(theo. sharp)" for the corner at \$(-15, 110)\$, followed by the coordinates \$(-15, -115.402729)\$. An arrow connects the initial point's path to this final shape.

$$V_{step} = \pi(|x_i| + |x_d|)(R_{step}^2 - R_i^2) \quad (11)$$
$$V_{art} = V_{outer} - V_{inner} - V_{step} - V_{rounds} \quad (12).$$

The symmetric nature of the artifact allows for the use of well-established volume estimation methods where a 2D-profile is revolved around a central axis (i.e. disc integration). From integral calculus, the definition is shown below in Eq. 13,

To define the function $f(x)$, the inner and outer profiles are modeled with respect to the analytical parameters shown in FIGURE 2 and FIGURE 3. For the inner profile, a simple piecewise function of x is determined in Eq. 14,

$$f_{out}(x) = \begin{cases} y_{sharp} + (\Delta y' - y_{sharp}) \frac{x}{h_{c_2}}, & 0 < x < 52.17 \\ \sqrt{R_o^2 - x^2}, & 52.176 < x < 130 \end{cases} \quad (15)$$

Closed Newton-Coates Formulae

Newton-Coates formulae are the preferred method when dealing with quadrature, particularly equally-spaced points over the interval $[a, b]$. Since the end points are also taken into account, the “closed” Newton-Coates formulae is preferred. The general formula for degree n is shown in Eq. 16,

$$\int_a^b f(x) dx \approx \sum_{i=0}^n w_i f(x_i) \quad (16)$$

where $x_i = hi + x_o$, $h = (b - a)/n$ and w_i as the weights. The first two degrees, the trapezoidal and the composite Simpson's rules will be the algorithms investigated. The reason for the composite Simpson's rule is to assess the possibility of non-smoothness of the function (Eq.s 14 and 15) over the interval $[a, b]$, therefore, breaking the interval into sub-intervals of different length to minimize the effects where the integrand is less well-behaved [5].

To model the Eq. 16 for degree $n = 1$, the Trapezoidal rule, one gets Eq. 17,

$$\int_a^b f(x) dx \approx \frac{h}{2} \sum_{i=0}^n [f(x_{i+1}) + f(x_i)] + E(x) \quad (17)$$

where the error function is defined as $E(x) = -\frac{h^3}{12} f^{(2)}(\xi)$. The function $f^{(2)}(\xi)$ is assumed continuous over the interval $[a, b]$ for some value ξ . For $n = 2$, composite Simpson's rule, one gets Eq. 18,

$$\int_a^b f(x) dx \approx \frac{h}{3} \sum_{i=0}^{n/2} [f(x_{2i-2}) + 4f(x_{2i-1}) + f(x_{2i})] + E(x) \quad (18)$$

where the error function is defined (and bounded) as $E(x) = -\frac{(b-a)h^4}{180} |f^{(4)}(\xi)|$.

Spline Curve

Historically at LANL, spline interpolation has been used to determine part definition for the various geometries using commercially-available CAD packages [9]. The methods previously mentioned need continuously differentiable (smooth), well-conditioned functions over the interval of interest for errors in the measurements to be at a minimum. Additionally, even-spaced intervals are necessary for the methods to work.

Mathematical spline curves approximate by a piecewise polynomial, usually a unique polynomial for each different arc between pairs of points. A natural cubic spline is used, where a

cubic polynomial is used between each pair of points. The general expression for a cubic spline is detailed below in Eq. 19,

$$f(x) = \begin{cases} S_0(x), & t_o \leq x \leq t_1 \\ \vdots & \\ S_{n-1}(x), & t_{n-1} \leq x \leq t_n \end{cases} \quad (19)$$

where the functions of degree 3 are defined as $S_0(x) = a_0x^3 + b_0x^2 + c_0x + d_0$ and $S_{n-1}(x) = a_{n-1}x^3 + b_{n-1}x^2 + c_{n-1}x + d_{n-1}$ over each respective subinterval. Furthermore, the first and second derivatives of $S(x)$ are continuous over the open interval $(-\infty, +\infty)$. Therefore, the volume is then estimated using Eq. 13. Error analysis for cubic splines can be a challenge on its own as the error is typically bounded based on the type of end conditions but is typically 60% of the global error for Closed Newton-Coates degree one (Trapezoidal) and around 0.5% greater than degree two (composite Simpson's) [10].

UNCERTAINTY EVALUATION

Estimating the measurement uncertainty associated with the calculation of the artifact density is done via the Law of Uncertainty Propagation (LUP) via the GUM principles [11]. The mathematical model of the measurement was determined as Eq. 1. Since both the mass and volume are determined using *different* experiments (i.e. mass balance measurement and CMM measurement data, respectively), the covariance between the two quantities is treated as zero; therefore the quantities are independent.

Applying the LUP to Eq. 1 yields the combined standard uncertainty, Eq. 20,

$$u(\rho) = \sqrt{\left(\frac{1}{V}\right)^2 u^2(m) + \left(-\frac{m}{V^2}\right)^2 u^2(V)} \quad (20)$$

where the m is the measured mass (in g) of the artifact, $u(m)$ is the standard uncertainty of the mass measurement (in g), V is the estimated volume of the artifact (in cm^3) and $u(V)$ is the standard uncertainty of the estimated volume. Assuming a confidence interval of 95% for the density results, the combined standard uncertainty is multiplied by a coverage factor of $k = 2$ to yield an expanded uncertainty of $U_{95} = k \cdot u(\rho)$.

EXPERIMENTAL APPROACH

The artifact measured is made out of 6061 aluminum which has a theoretical density of 2.71 g/cm^3 [12]. For the mass measurements, a calibrated mass-balance (accuracy of $\pm 0.005 \text{ g}$) is used to collect three measurements over the course of a week in temperature-controlled lab. Upon completion of the measurement process, the artifact mass value was determined to be $(1435.230 \pm 0.070) \text{ g}$.



FIGURE 5. Measurement setup of the artifact on the Global Image CMM and fixture.

The measurement data of the artifact used to estimate the volume was collected using a Brown & Sharpe Global Image CMM, shown above in FIGURE 5. Both the inner and outer surfaces of the artifact were measured using continuous scanning to collect high-data density measurement sets to cover as much of the artifact's surface area as shown in FIGURE 6. A total of 240 scans were collected where the average (2D profile) was used for volume analysis.



FIGURE 6. Artifact measurements: (A) inner contour; and (B) outer contour.

The artifact was measured three times over the course of a week, as well as in different positions throughout the measurement volume, to capture the effects from extrinsic error sources. The measurements, CMM and mass, were collected in environmentally-similar labs, where the temperature was certified at $(20.0 \pm 1.0)^\circ\text{C}$. Upon completion of the measurements, the volume (and density) estimations were numerically calculated and the results are shown in TABLE 1.

TABLE 1. Volume and density estimation results for the unclassified CMM artifact.

Method	Volume (cm^3)	Density (g/cm^3)
Analytical*	534.966 ± 0.000	2.683 ± 0.000
Trapezoidal	538.054 ± 2.645	2.667 ± 0.013
Simpson's	537.941 ± 1.613	2.668 ± 0.008
Spline	535.536 ± 1.902	2.679 ± 0.010
Measured**	531.960 ± 0.000	2.698 ± 0.001

*Contribution from mass to density uncertainty is negligible.

**Value was determined using Archimedes Principle and no uncertainty was given for volume measurement.

The results in TABLE 1 show the numerical methods agree reasonably well, with the spline integration closes to the analytical results. Moreover, all four methods show larger volume results, thus smaller density estimates, than the measurement of the artifact when using Archimedes Principle. This is likely the result of using the average of the 240 scans for the estimations instead of piecing together each individual volume slice. However, the density estimations were all still within the tolerance of $\pm 0.01 \text{ g/cm}^3$ from the measured result.

CONCLUSIONS AND FUTURE WORK

A method for estimating workpiece density using known numerical techniques and quadrature was presented. The results indicated that it is possible to estimate the workpiece density results, using measurement data collected with a CMM, confidently with small uncertainty. However, this is only but a conservative estimate at best since the average of the 240 scans is used. Surface defects on the artifact are averaged-out and minimal information about the defects is gained.

To combat this issue, a 3D approach to estimating the artifact volume will be investigated further. The initial thought is the estimate the volume between each of the 240 scans as “wedges” of the artifact and determining a volume estimate by summing up all the calculated “wedges” for a more complete assessment.

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